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Investigations on pump running in turbine mode: A review of the state-of-the-art

Sanjay V. Jain*, Rajesh N. Patel

Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad 382481, India

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ABSTRACT

In remote communities where it is not economical and practically possible to take the grid connection, stand-alone small hydro systems can be used to fulfill the energy requirement. Small-scale hydroelectric power systems are emerging as a promising source of renewable energy generation, but they require low cost hydraulic and electric equipments to make them economically feasible. In such plants, pumps can be used in turbine mode considering various advantages associated with pump e.g. ease of availability, proven technology, low initial and maintenance cost, availability for a wide range of heads and flows, etc. The efficiency of pump as turbine (PAT) is usually lower than that of conventional hydro turbines. However, efficiency is not the primary selection criterion for such machines and it is recommended to operate such machines around maximum efficiency point.

In the present study, different turbines suitable for micro-hydropower plants are discussed. The historical development of PAT is described. The review of the state-of-the-art of pump running in turbine mode is presented. Different pumps suitable to run in turbine mode for low capacity power generation in micro-hydropower plants as well as in water supply piping systems are discussed. Theoretical, experimental and numerical investigations carried out by different researchers on PAT are reviewed. The research work on PAT including criteria for selection of pump running as turbine, cavitation analysis, force analysis, loss distribution, various methods of performance enhancement, cost analysis of hydropower plant with conventional hydro turbine and PAT, applications of PAT in water supply pipelines, etc. is discussed. The worldwide implementation of PAT and different manufacturers of PAT are described. The limitations in implementation of PAT as well as the recommendations to improve the performance of PAT are described. The current trends and future scope for the further improvement and implementation of PAT are also discussed.

Contents

* Corresponding author. Tel.: +91 9998623087.
E-mail addresses: sanjay.jain@nirmauni.ac.in, svjain5@yahoo.com (S.V. Jain).

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Nomenclature

H	head (m)
Q	discharge (m^3/s)
N, n	rotational speed (rpm)
N_s	specific speed
C	prediction coefficient
V	absolute velocity (m/s)
W	relative velocity (m/s)
D	impeller diameter (m)
P	power (W)
Qu	unit discharge
Nu	unit speed
h	head correction factor
q	discharge correction factor
f	frequency (Hz)
p	number of poles
g	gravitational constant (m/s^2)
t	time (s)
d	annual discount rate (%)
PAT	pump as turbine
MW	megawatt
kW	kilowatt
SHP	small hydropower
MNRE	ministry of new renewable energy
BEP	best efficiency point
ISO	international organization for standardization
ANN	artificial neural network
MATLAB	matrix laboratory
CFD	computational fluid dynamics
CSHN	combined suction head number
FEM	finite element method
RANS	Reynolds-averaged Navier–Stokes
FVM	finite volume method
MRF	moving reference frame
RNG	renormalization group
SST	shear stress transport
RAM	random-access memory
SIMPLE	semi-implicit method for pressure-linked equations
SIMPLEC	semi-implicit method for pressure-linked equations-consistent
PISO	pressure-implicit with splitting of operators
PRV	pressure reducing valve
UNIDO	United Nations industrial development organization
ETC	environmental tectonics corporation
TaTEDO	Tanzania traditional energy development organization
ESP	engineering studies program
BGET	border green energy team
PVC	polyvinyl chloride
IG	induction generator
NPV	net present value
BCR	benefit/cost ratio
IRR	internal rate of return
ALCC	annual life cycle cost
US$	United States dollar
Wp	watts peak
Ah	ampere hour
Co	initial cost
CRF	capital recovery factor
Ac	annual expenses
L	equipment life
PV	photovoltaic
EGE	energy generation equipment
CW	civil works
IGC	induction generator controller
KBL	Kirloskar brothers limited
rpm	revolution per minute
bhp	brake horse power
ft	foot
usgpm	United States gallons per minute
lps	liter per second
kWh	kilowatt hour

Greek symbols

η	efficiency
γ	specific weight (N/m^3)
ψ	head coefficient
ϕ	discharge coefficient
π	power coefficient
χ	relation between best efficiency and specific speed of pump
ρ	density of water (kg/m^3)
μ	blade torque coefficient
σ	Thoma's cavitation coefficient/slip factor/axial force coefficient
k	turbulence kinetic energy (J/kg)
ε	turbulence dissipation rate (J/kg-s)

Subscripts

t, T	turbine
p	pump/peak
h	hydraulic
g	generator
m	motor/mechanical
2u	tangential component at pump outlet/turbine inlet
2m	meridian component at pump outlet/turbine inlet
n	net
v	volute
l	leakage
e	kinetic energy
i	hydraulic
crit	critical
opt	optimum

1. Introduction

Hydropower is probably the oldest renewable energy source known to the mankind for mechanical energy conversion as well as electricity generation. It is a renewable, non-polluting and environment friendly source of energy. It is inflation free source due to absence of fuel cost with mature technology along with highest prime mover efficiency and enormous operational flexibility. Hydropower contributes to around 16% of the World electricity supply generated from about 20,053 TWh of installed capacity [1] and in many countries it is the main source of power generation e.g. Norway – 99%, Brazil – 86%, Switzerland – 76% and Sweden – 50% [2].

The history of evolution and growth of renewable energy sources in India is as old as the history of power development and goes back to about 110 years. The first hydropower plant of 130 kW capacity was installed in 1897 at Sidrapong, Darjeeling which was Run-off-river type [3]. The need of power generation from renewable energy sources was not much earlier. The interests in renewables started much later in 1970s when the quest for alternative energy sources began after the first oil shock [4]. Promoting renewable energy in India has assumed great importance in recent years in view of high growth rate of energy consumption, high share of coal in domestic energy demand, heavy dependence on imports for meeting demands for petroleum fuels and volatility of world oil market [5].

Although, hydropower projects being considered as the most economic and preferred source of electricity, the share of hydropower in India has been declining since 1963. The hydro share decreased from 50% in 1963 to 26% in 2005, which has been further reduced to about 18% in 2013. For grid stability the ideal hydro-thermal mix ratio is 40:60 which is required to be achieved in the near future to meet the grid requirements and peak power shortage [2,6]. The total hydroelectric power potential in the country is about 150,000 MW, equivalent to 84,000 MW at 60% load factor [7].

While wind and solar power are getting most of the hype due to green energy being such a hot topic these days, hydropower, the biggest and oldest source of renewable energy in the world, is getting another look in less obvious areas [8]. A huge number of large hydropower generation schemes have been implemented and the technology is now quite mature; however, large hydro suffer from problems like large submergence of valuable land, resettlement and rehabilitation of affected people, long gestation and payback periods, large gaps between survey and reality, as well as political issues [9]. Hence, the possibility for the implementation of large hydropower plants, though advantageous from the economies of scale but may not always be technologically, economically, socially, politically or environmentally feasible. Consequently, interest in small scale hydroelectric power generation has increased rapidly in last decade.

Recognizing the fact that, small hydropower (SHP) projects can provide a solution for the energy problem in rural, remote and hilly areas, where extension of grid is techno-economically not feasible, as well as along the canal systems promoting small hydropower projects is one of the objectives of the Policy on Hydropower Development in India [2]. SHP projects became attractive after the oil price crisis of the 70s and again in recent years [10]. In India, there is a huge network of irrigation canals on which number of falls are available where energy is just wasted in the formation of hydraulic jumps. SHP stations can be installed at canal falls as well as at dam toe in addition to natural falls available.

Looking to the benefits of small hydro and necessity for its development, Government of India has increased the limit of small hydro up to 25 MW from 15 MW to include more projects under this category and many financial and administrative incentives are being offered to developers of small hydro [11]. The SHP is further categorized as micro (up to 100 kW), mini (101–2000 kW) and small hydropower (2001–25,000 kW) [12]. India has an estimated SHP potential of about 20,000 MW; of this 19,749 MW has been identified from 6474 potential sites and 967 projects with an aggregate capacity of 3632 MW have already been installed. The Ministry of New Renewable Energy (MNRE), Government of India decided that out of the total grid interactive power generation capacity that is being installed, 2% should come from small hydro. This translates to about 2100 MW capacity addition during 2012–2017 [6].

Micro-hydropower can be one of the most valuable answers to the question of how to offer isolated rural communities the advantages of electrification and the associated progress, as well as to improve the quality of life. It is one of the most cost-effective energy technologies for rural electrification in less developed countries [13]. The MNRE has identified more than 6000 streams in north and north-east regions of India; which are not suitable for installation of large hydropower plants but are favorable for electricity generation in the range of 5–100 kW. Small, mini- and micro-hydropower schemes can be installed on such water streams [14]. The present focus of MNRE for the SHP development is to lower the cost of equipment, increase its reliability and set up projects in areas that give the maximum advantage in terms of capacity utilization [6].

The problems associated with small hydropower development are technical and economical as compared to conventional

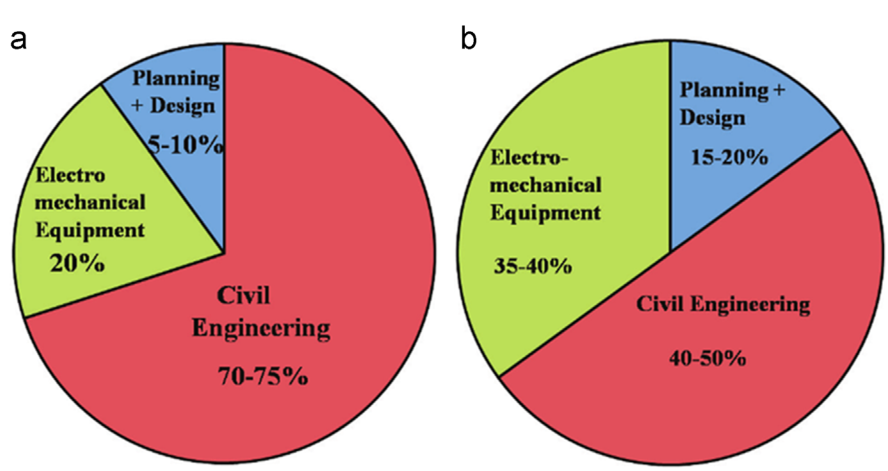

Fig. 1. Cost distribution of (a) large and (b) micro-hydropower plants.

hydropower plants. The main limitation in installation of mini/micro-hydropower plants is higher cost of hydro turbines as the turbine manufactures do not manufacture the turbines in smaller capacities. Also, it is very difficult, time consuming and costly to develop site specific turbines correspond to local ecology in low capacity range. The cost details of different equipments for large and micro-hydropower projects are given in Fig. 1. The cost of electro-mechanical components in large hydro is around 20% but in micro-hydro it is relatively high and varies from 35% to 40% of the total project cost [15]. This cost may rise even up to 60% to 70% of the total project cost in some typical cases [11]. Hence, by decreasing the cost of electro-mechanical equipments micro-hydro schemes may become more favorable and easily accessible.

Many investigators have attempted different turbines in small/mini/micro-hydro range viz. cross flow turbine, Turgo turbine, single and multi-jet Pelton turbines, Francis turbine but their experience was not much encouraging. Williams [16] reported that in micro-hydro range, a cross flow turbine requires a large turbine running at slower speed, resulting in need of a belt drive to power a standard generator. A Pelton turbine for such applications would require three or four jets, resulting in a complicated arrangement for the casing and nozzles. A small Francis turbine could also be used in this range, but it would be even more expensive than a cross flow turbine.

In any water system which has excessive available energy e.g. natural falls, water supply, irrigation, sewage or rain systems, the application of a pump instead of a turbine seems to be an alternative solution in terms of easy implementation and reduced cost of equipments [17]. The pumps in turbine mode have various advantages over conventional hydro turbines viz. low cost, less complexity, mass production, availability for a wide range of heads and flows, short delivery time, availability in a large number of standard sizes, ease of availability of spare parts, easy installation, etc. [18,19]. For low and very low capacity power plants (upto500 kW), the PAT deserves thoughtfulness due to considerable reduction in the capital cost of the plant of the order of 10–1 or even more in spite of having slightly reduced efficiency [20]. In this range, the investment cost for conventional hydro turbines is relatively high. The payback period of such hydro turbine can be as high as 15 years which can be reduced to 2 years using PAT for similar capacity [21,22].

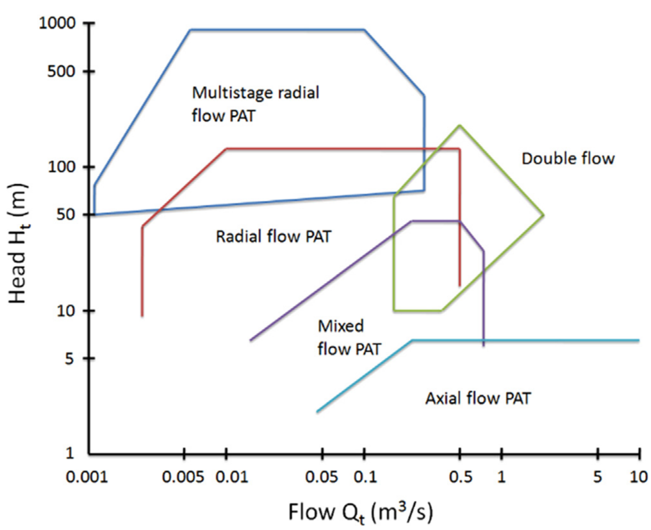

Fig. 2. Different pumps suitable as turbines.

2. Historical development of PAT

When the pumps were first used in turbine mode is unclear. In 1931, when Thoma and Kittredge [23] were trying to assess the complete characteristics of pumps, they accidentally realized that pumps could operate very efficiently in the turbine mode. Later in 1941, Knapp [24] published the complete pump characteristics for a few pump designs based on experimental investigations. In 1950s and 1960s, the concept of pumped storage power plants, in the range of 50–100 MW, was evolved mainly in developed countries to satisfy the peak power demands. In later years, chemical industries became another area for the application of PATs for energy recovery. Even in water supply networks applications of this technology were found. This background gave some momentum to a rich phase of research and then onwards, standard manufactured pumps were studied in turbine mode. In later years, many other techniques were developed by lots of researchers [14].

The technology for the use of PAT for electrical power generation was not available earlier. However, advances in electrical machinery control technologies, rotation sense and torque have created the possibility of the utilization of pump rotating in reverse mode for power generation [20]. Agostinelli and Shafer [25] tested many pumps in turbine mode over the years and concluded that when a pump operates in a turbine mode, its mechanical operation is smooth and quiet; its peak efficiency is same as in pump mode; head and flow at the best efficiency point (BEP) are higher than that in pump mode and the power output is higher than that the pump input power at its best efficiency.

3. Selection of pump running in turbine mode

Selection of proper pump to be used in turbine mode for a particular site is a big issue in installation of PAT. The different factors for selecting reverse running pumps can be summarized in terms of available head range, capacity range, back pressure at turbine outlet, desired speed, etc. [26]. For the cost-effective installation of PAT, it is essential to select proper type of pump. It is also required to predict its performance in turbine mode in advance before its installation. Many researchers have done lots of studies on these issues and proposed various correlations for the selection of pump which are summarized in this section.

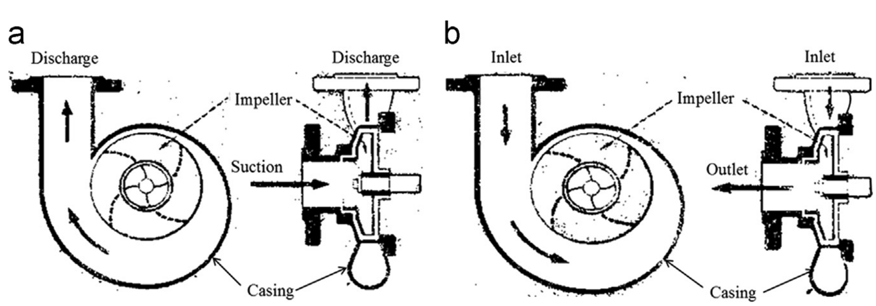

Fig. 3. Centrifugal pump in (a) pump and (b) turbine modes.

Table 1
Historical development in the performance prediction methods of PAT.

Year	Name of investigator	Criteria	Head correction factor (h)	Discharge correction factor (q)	Remarks
1957	Stepanoff [32]	BEP	$\frac{1}{\eta_p}$	$\frac{1}{\sqrt{\eta_p}}$	Accurate for N_s: 40–60
1962	Childs [33]	BEP	$\frac{1}{\eta_p}$	$\frac{1}{\eta_p}$	–
1963	Hancock [34]	BEP	$\frac{1}{\eta_t}$	$\frac{1}{\eta_t}$	–
1980	Grover [41]	Specific speed	$2.693-0.0229N_{st}$	$2.379-0.0264N_{st}$	Applied for N_s: 10–50
1982	Hergt [from 42]	Specific speed	$1.3-\frac{6}{N_{st}-3}$	$1.3-\frac{1.6}{N_{st}-5}$	–
1985	Sharma [36]	BEP	$\frac{1}{\eta_p{}^{1.2}}$	$\frac{1}{\eta_p{}^{0.8}}$	Accurate for N_s: 40–60
1988	Schmiedl [37]	BEP	$-1.4+\frac{2.5}{\eta_{hp}}$	$-1.5+\frac{2.4}{\eta_{hp}{}^2}$	–
1994	Alatorre-Frenk [38]	BEP	$\frac{1}{0.85\eta_p{}^5+0.385}$	$\frac{0.85\eta_p{}^5+0.385}{2\eta_p{}^{9.5}+0.205}$	–
1998	Sharma [28]	BEP	$\left[\frac{N_g}{N_m}\right]^2 \times \frac{1.1}{\eta_p{}^{1.2}}$	$\frac{N_g}{N_m} \times \frac{1.1}{\eta_p{}^{0.8}}$	$N_g = 240 \times \frac{f}{p} - N$

3.1. *Types of pumps suitable as turbines*

Selection of appropriate type of pump depends on head and discharge available at the site, initial and maintenance cost, ease of availability of pump, etc. Lueneburg and Nelson [26] reported that all centrifugal pumps from low to high specific speed, single or multistage, radially or axially split, horizontal or vertical installations can be used in reverse mode. The pump to be used as turbine can be selected, based on head and discharge, as shown in Fig. 2 (adapted from Ref. [27]). It can be seen that multistage radial flow pumps are suitable for high head and low discharge sites; whereas, axial flow pumps are appropriate in low head and high discharge range. Similar chart was also presented by Orchard and Klos [21] (range: 5–750 kW).

The field applications of multistage, single impeller centrifugal and axial flow pumps can be compared with Pelton, Francis and Kaplan turbines respectively [10]. It is possible to use in-line and double suction pumps in turbine mode; but they are less efficient in turbine mode. Self-priming pumps cannot be used in turbine mode due to the presence of non-return valve. Dry-motor submersible pumps containing fin cooling arrangement for motor, are also not suitable for turbine mode operation due to overheating issue. Wet-motor submersible bore hole pumps usually contain a non-return valve and may also support thrust bearing; hence, are normally not suitable for turbine mode applications [28]. Pump works as turbine most efficiently in the head range of 13–75 m. Also, as head increases the cost per kW decreases [29]. The working principles of end suction, single stage centrifugal pump in pump and turbine modes are shown in Fig. 3 [30]. It can be seen that, in turbine mode flow direction is reversed as compared to pump mode.

3.2. *Criteria for performance prediction of pump running in turbine mode*

PAT can become a cost-effective alternative to traditional turbines as long as the turbine mode performance can be predicted before their installation [31]. Normally, the pump manufacturers do not offer performance curves of their pumps in turbine mode; which makes it difficult to select a suitable pump to run as a turbine for a particular operating condition. One of the main objectives of all PAT researchers all over the world has been to build a method that would make accurate predictions of the turbine operation of pumps.

A large number of theoretical and experimental studies have been found in the literature for the prediction of performance of PAT in which relations between BEPs in pump and turbine mode were derived based on either efficiency or specific speed in pump mode. The specific speed is one of the main parameter of turbomachines that characterizes the type of the runner (i.e. radial, mixed or axial flow), blade shape, spiral casing and other design features. The BEP parameters in turbine mode are different from that in pump mode. Hence, the relation between these parameters is derived by the investigators in terms of head and discharge correction factors (h and q), which are defined as under:

$$h = \frac{H_t}{H_p} \tag{1}$$

$$q = \frac{Q_t}{Q_p} \tag{2}$$

Stepanoff [32], Childs [33], Hancock [34], McCluskey and Lundquist [35], Sharma [36], Luneburg and Nelson [26], Schmiedl [37] and Alatorre-Frenk [38] developed relations based on efficiency in pump mode; whereas, Kittredge [39], Diederich [40], Grover [41], Lewinski & Keslitz [42], Hergt (adapted from Ref. [42]) and Buse [30] derived the relations based on the specific speed of the pump.

The historical development in the performance prediction methods of PAT (up to late 90s) is presented in Table 1, which shows the equations for h and q derived by different researchers. These methods produce a wide range of results and the deviations between performance predicted by these methods and experimental results have been found to be around $\pm 20\%$ or even more [27,16]. Therefore, these methods are confined to preliminary selection of pumps to be used as turbines which is important to obtain a rough estimation of turbine mode characteristics from pump mode characteristics [43]. Few of these methods may not include the performance prediction at no-load, part load and overload operating conditions. Further, these methods have not been tested on different pump shapes to get more insights about their features like reliability, accuracy and robustness [44].

Table 2
Comparison of turbine prediction methods [50].

Prediction method	Mean value of C	No. of pumps for which $C > 1$	Pumps out of range (%)
Schmiedl [27]	1.173	13	40
Stepanoff [32]	0.847	12	34
Childs [33]	0.921	14	40
Hancock [34]	0.906	10	32
Sharma [36]	0.733	7	20
Alatorre-Frenk [38]	0.852	10	29
Grover [41]	1.333	22	81
Hergt [42]	0.865	11	32

Few other researchers like Knapp [24], Wong [45], Cohrs [46] and Amelio and Barbarelli [47] have also given the PAT prediction

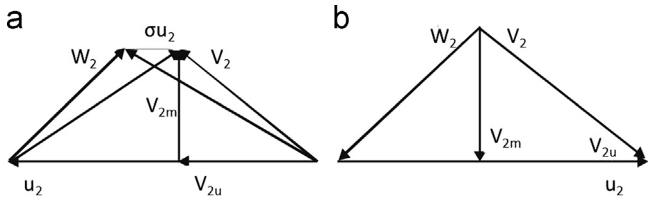

Fig. 4. Velocity triangles at (a) pump outlet and (b) turbine inlet.

methods based on the pump design, its geometry and assumptions of some complex hydraulic phenomena like losses and slip effects to come up with more accurate method. These methods are quite comprehensive but they are difficult to put in practice and outside the reach of planners, since these methods necessitate very detailed information, which is sometimes patented or available only with the manufacturers [48]. In last few years, many researchers have used different theoretical and experimental techniques to predict the PAT performance from pump characteristics which are described in this section.

3.2.1. *Theoretical studies for performance prediction*

Different methods for predicting PAT performance have been proposed in the literature; some of them are based on pump-mode performance and other depends on the geometry of the machine. By means of the economic methodology, the accuracy of the former methods was evaluated by Alatorre-Frenk [38]. A new method based on pump-mode performance and type of casing was developed as illustrated in Table 1. It was found that, the inaccuracy of the prediction method reduces the benefit-to-cost ratio of the scheme by less than 1% which may be considered negligible in view of the high cost of turbine-mode laboratory tests which would otherwise be necessary.

Sharma [28] discussed the suitability of different pumps which can be used as turbines. The analysis of performance curves in pump and turbine mode plotted by Grant and Bain [49] revealed that the location of turbine BEP is at a higher flow and head than the pump BEP. The equations for h and q were presented to predict turbine performance from pump data, as illustrated in Table 1, where N_m and N_g are the motor and generator speeds in rpm; η_p and η_t are the efficiencies in pump and turbine mode; f is the frequency and p is the number of poles. It was pointed out that the proposed equations were only approximate and the actual values of Q_t and H_t might vary by $+20\%$ of the predicted value for the BEP. Hence, after initial selection, it was recommended to test the pump under turbine condition to find out power at the available head. The advancements in pump manufacturing technology and various ways of reducing the cost of PAT based hydropower plants were also discussed.

Williams [50] compared eight different PAT performance prediction methods based on turbine tests on 35 pumps in the specific speed range of 12.7–183.3 and studied the effects of poor turbine prediction on the operation of a PAT. The difference between the predicted BEP and the actual BEP for the PAT was studied using the prediction coefficient (C). The value of C was computed from the ellipse formed on head versus discharge (H–Q) chart centered on the guaranteed BEP as per the British Standard and the criterion for an acceptable prediction was $C \le 1$. The mean value of C and the number of pumps for which the value of C was greater than 1 (i.e. the error is outside the acceptable limits) for different methods is listed in Table 2. None of the eight methods had given an accurate prediction for all the pumps but Sharma's method was found to be better than the other methods; hence, it was recommended as a first estimate for prediction of the turbine performance. It was suggested that whenever possible; once a pump is selected for a micro-hydro scheme it shall be tested in turbine mode before its installation on the site.

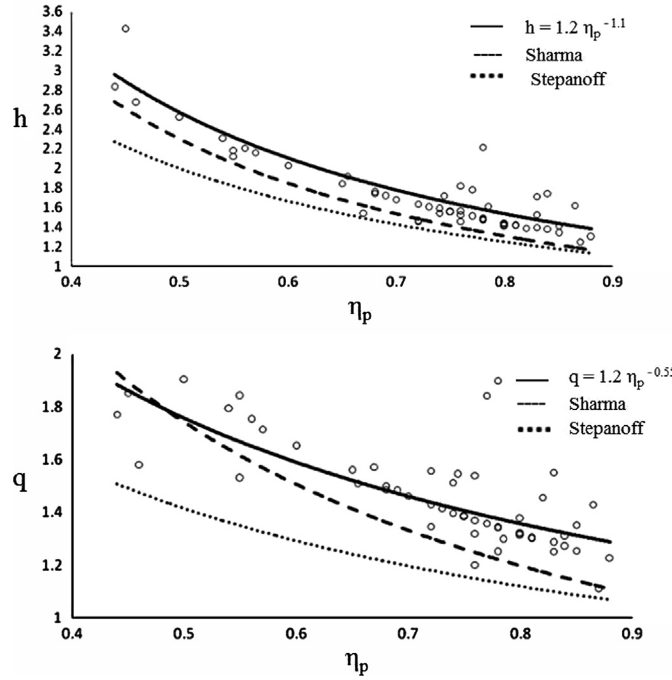

Fig. 5. (a) Head ratio and (b) discharge ratio for tested PATs with various pump maximum efficiencies.

Saini [51] developed nomogram based on the specific speed in pump mode which can be used to determine the correction factors/conversion coefficients for head and discharge. Using these factors, the values of the BEP parameters in turbine mode can be directly achieved from the nomogram which reduces the tedious calculations involved in the selection.

Fernandez et al. [20] emphasized that majority of the analysis carried out for performance prediction was based on the hypothesis of the similarity between maximum efficiencies in pump and turbine modes, which is not easy to maintain, and other studies were on the basis of algebraic relations as a function of efficiency. Further, all these studies were done considering equal rotational speeds in both the modes. In this paper, the characteristics of pump running in reverse mode at different rotational speeds were presented. The experiments were performed on a hydraulic set-up developed according to ISO 3555:1977 in the pump and turbine modes; and constant speed and constant head characteristics were drawn. The fundamental equation of Euler's head was modified by applying hydraulic characteristics of pump and turbine like hydraulic efficiency, slip factor, etc. and an equation for performance prediction of pump in turbine mode was derived. The velocity triangles at outlet of pump and inlet of turbine are shown in Fig. 4.

Joshi et al. [52] established the relationship between pump and turbine specific speeds (N_{sp} and N_{st}), as given below, for selection of PAT particularly for low head sites. The constant speed and constant head characteristics for pump and PAT were plotted in terms of normalized head, power and efficiency as a function of normalized discharge for an axial flow pump using the data from Stepanoff [32]. To select the pump which can be used as turbine, the value of specific speed in turbine mode can be worked out and based on that the pump specific speed can be read from $N_{st}-N_{sp}$ plot. The approach had the advantage of simplicity and generality but it was based on experimental data from only 3 pumps and the effect of efficiency was not considered; hence, the method was

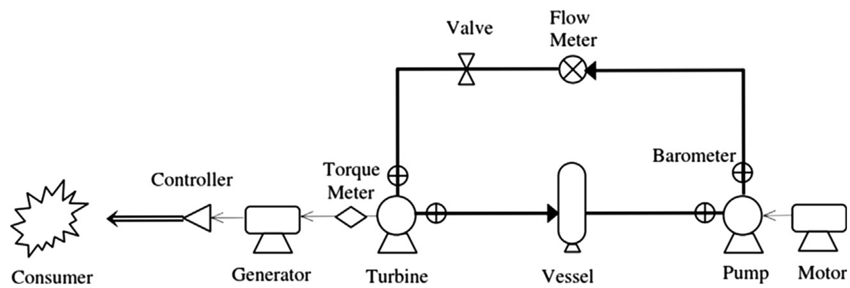

Fig. 6. The mini-hydropower test rig established in University of Tehran.

recommended for the approximate analysis only.

$$\frac{N_{st}}{N_{sp}} = \frac{(\text{rpm} \times \sqrt{\text{bhp}}/\text{ft}^{5/4})_t}{(\text{rpm} \times \sqrt{\text{usgpm}}/\text{ft}^{3/4})_p} \tag{3}$$

Isbasoiu et al. [53] mentioned that PAT selection must be done based on the availability of head and discharge for a specific site. Usually, the discharge is decided on the basis of the minimum flow rate and the head is evaluated after subtracting the head loss in the penstock from the elevation difference between intake and turbine outlet. The chosen pump should have the head and flow, at the BEP, as far as possible similar to the site requirements. Although the PAT efficiency will be approximately same as in pump mode, at BEP the head and discharge values are quite different in pump and turbine modes. To achieve the reliable performance of a PAT, it was recommended to test it in the wide range of head and discharge. In addition, for the selection of the feed pump, it was pointed out that the feed pump must be able to generate higher head and discharge than that required at the BEP of PAT and its power rating should be around 4 times than that of PAT.

Pandey and Saini [54] developed an Artificial Neural Network (ANN) based model using their own experimental data to predict the performance of PAT. NNtool-box (MATLAB) software was used for the performance analysis of PAT and curves were developed for the power output and efficiency of PAT. The model was used to train, validate and test the data at different head and flow rates ranging from 7 to 21 m and 24.5 to 50 lps respectively. Head, flow rate and power input were taken as input parameters and power output and efficiency of PAT were considered as the target data for the model. A comparison between the ANN output and actual experimental data was made to validate the ANN model. It was emphasized that the proposed method might be helpful for evaluating the performance of PAT at different heads and flow rates and hence the ANN model may be considered a good tool for selection of pump in turbine mode.

Singh and Nestmann [48] presented an optimization routine for prediction (predicting performance of pump in turbine mode) and selection (selecting the most appropriate pump for turbine-mode operation) of radial flow centrifugal pumps in low specific speed range. The approach was based on experimental results of 9 models of PAT (N_{sp}: 20–80 rpm) and the fundamentals of applied turbomachinery like the specific speed–specific diameter plot (called the Cordier/Balje plot), which was first introduced by Cordier [55] and then followed intensively by Balje [56]. The method was experimentally evaluated for three pumps with specific speeds of 18.2 rpm, 19.7 rpm and 44.7 rpm, and the error in turbine performance prediction was found to be within $\pm 4\%$ in the full load operating range. It was reported that the application of the proposed model was quite simple and could be incorporated in any computer program which could be made accessible to small manufacturers and system planners particularly for energy recovery or micro-hydro projects [57].

Yang et al. [22] developed a theoretical method to envisage performance of PAT based on former research results, theoretical analysis and empirical correlation given below. The effects of variations of specific speed and maximum efficiency of pump on h and q were studied and found that two pumps with same specific speeds may have different h and q. Consequently, a pump was simulated in pump and turbine modes using commercial 3D Navier–Stokes Computational Fluid Dynamics (CFD) code available in ANSYS-CFX which has utilized a finite-element based finite-volume technique for discretization of the transport equations. The comparison of the proposed method with other two methods viz. Stepanoff [32] and Sharma [28], as shown in Fig. 5, revealed that BEP parameters obtained with the proposed method and CFD analysis were found to be in close resemblance compared to other two methods. The minor variation among experimental and numerical values may be attributed to the negligence of leakage loss through balancing holes, mechanical loss caused by mechanical seal and bearings and the surface roughness value set on the machine's surface.

$$h = \frac{1.2}{\eta_p^{1.1}} \tag{4}$$

$$q = \frac{1.2}{\eta_p^{0.55}} \tag{5}$$

3.2.2. Experimental studies for performance prediction

Derakhshan and Nourbakhsh [10] derived some relations to predict BEP of PATs based on the experimental work carried out on four centrifugal pumps with different specific speeds of 14.6, 23, 37.6 and 55.6 (m,m^3/s). At same rotational speed, the head and discharge values were found to be higher in turbine mode compared to that in pump mode; however, the efficiencies were observed to be almost same for both modes. Pumps with higher specific speeds were subjected to lower values of h and q. A new method was proposed to predict BEP of PAT based on hydraulic specifications of pump, particularly the specific speed, which represents the runner type and consequently its hydraulic behavior. It was shown that among the two pumps (having same specific speeds) the pump with bigger impeller works more efficiently. Also, the more efficient pump operates as turbine at larger values of h and q. The predicted h and q by this method were found in good agreement with the experimental results. A procedure to choose a proper centrifugal PAT for a small hydro-site with $N_{st} < 150$ (m,kW) based on h and q was presented as mentioned below.

$$h_{\text{new}} = h(0.25/D)^{1/4} \tag{6}$$

$$q_{\text{new}} = q(0.25/D)^{1/6} \tag{7}$$

Derakhshan and Nourbakhsh [58] predicted the BEP of centrifugal pump in turbine regime using theoretical analysis based on the “area ratio” method proposed by Williams [59] and

Table 3
Parameters affecting the performance prediction of PAT [43].

Parameters	Factors
Mechanical considerations	High horse power High suction pressure Operating speed Operating temperature Running clearances
Pump liquid	Slurries Abrasives High viscosity Dissolved gases
System considerations	Net positive suction head Suction and discharge piping arrangement Shape of head-discharge curve Run off conditions Vibration and noise limits

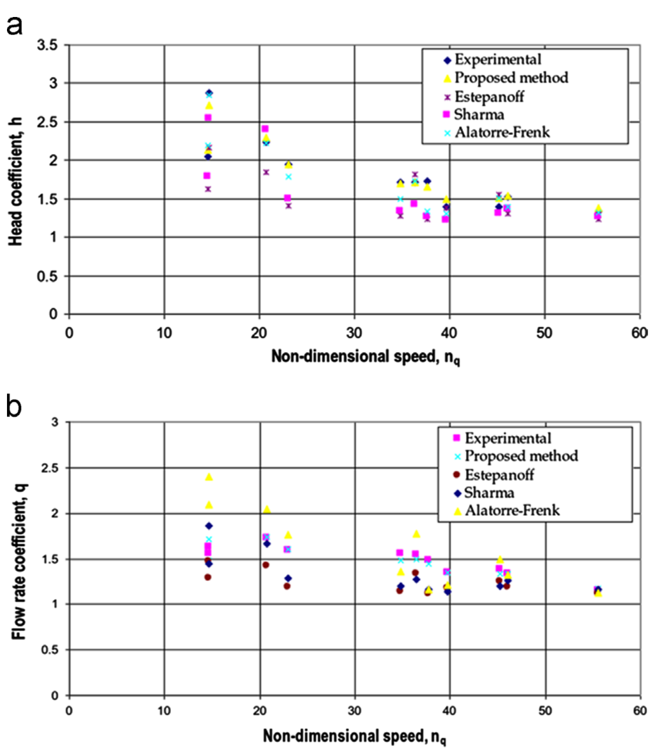

Fig. 7. Comparison of the proposed method with other methods for (a) h and (b) q.

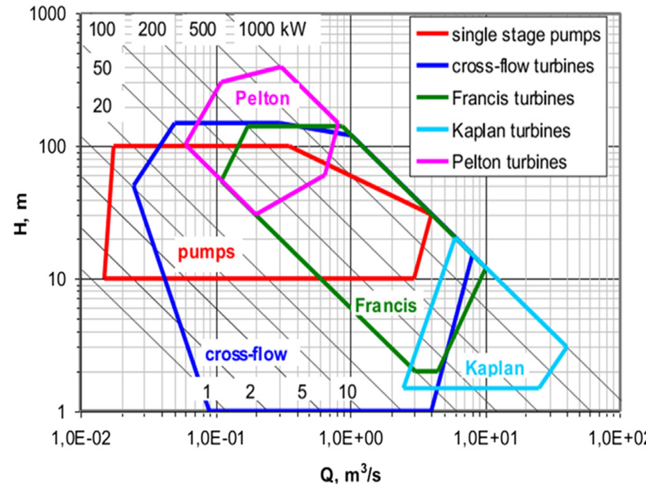

Fig. 8. Application range of conventional turbines and PAT.

Anderson [60]. The maximum efficiency of PAT was calculated as the ratio of net power output from the turbine and the hydraulic power supplied at the inlet. The net power output was worked out by subtracting various losses in the turbine (e.g. volute power losses, leakage losses, kinetic energy losses at the outlet, hydraulic losses and mechanical losses) from the gross power. A complete mini-hydropower test rig established in the laboratory of University of Tehran, shown in Fig. 6, was used for the experimental verification of theoretical results. At BEP, the values of discharge number, head number, power number and efficiency predicted by theoretical methods were found to be 1.1%, 4.7%, 5.25% and 2.1% lower than those of corresponding experimental values. These deviations might be due to assumptions made in the evaluation of the volute and the impeller losses. The equation of maximum efficiency of PAT was derived as:

$$\eta_t = \frac{P_{nt}}{\gamma \times Q_t \times H_t} = \frac{(\gamma \times Q_t \times H_t) - P_{vt} - P_{lt} - P_{et} - P_{it} - P_{mt}}{\gamma \times Q_t \times H_t} \tag{8}$$

Nautiyal et al. [43] developed relations using experimental data of the tested pump ($N_s = 18$ m, m^3/s) and pumps of some previous researchers to predict turbine mode performance from pump mode characteristics. The experimental results were presented in the form of non-dimensional parameters viz. head coefficient (ψ), discharge coefficient (ϕ) and power coefficient (π) which are expressed below. Various parameters affecting the performance prediction of PAT were enumerated as mentioned in Table 3. As compared to other methods, the deviation between experimental results and results obtained by proposed relations was less which made the performance prediction of PAT simpler and closer to accuracy. However, some uncertainties were still found in prediction of turbine mode characteristics using pump operation data. The relation (χ) between best efficiency and specific speed of pump was developed by regression analysis and equations of h and q were developed which are expressed below:

$$\psi = \frac{g \times H}{n^2 \times D^2}; \tag{9}$$

$$\phi = \frac{Q}{n \times D^3}; \tag{10}$$

$$\pi = \frac{P}{\rho \times n^3 \times D^5} \tag{11}$$

$$\chi = \frac{\eta_p - 0.212}{\ln(N_S)} \tag{12}$$

$$q = 30.303\chi - 3.424 \tag{13}$$

$$h = 41.667\chi - 5.042 \tag{14}$$

Engeda [61] and Florez and Jimenez [62] presented a comparison of the dimensionless coefficients (h and q) that relate the operating points of maximum performance in both modes (pump and turbine) and emphasized the necessity of more reliable and actual data for validation of prediction methods. Garcia et al. [63] compared and analyzed several prediction methods and algorithms suggested by Derakhshan and Nourbakhsh [10], Stepanoff [32], Sharma [36] and Alatorre-Frenk [38]. Fig. 7 shows the relationship between the dimensionless parameters of head and flow rate as pump and as turbine at the point of maximum efficiency. The head and flow rate were found to be higher in case of PAT but the efficiency was found to be almost equal in both the modes.

Agarwal [64] reported that conversion factors for PAT can be obtained on the basis of theoretical and numerical studies but the

performance of PAT cannot be anticipated perfectly. The need of further investigations to develop a common model for deriving the conversion factors was emphasized.

Many researchers have developed the charts for the selection of turbines (including PAT) in mini/micro-hydro range. The selection chart proposed by Engeda et al. [65] (range: 1–1000 kW) based on head and discharge is shown in Fig. 8. Williams [16] also published similar chart and reported that pumps can be used as turbine over the range normally covered by multi-jet Pelton turbines, cross flow turbines and small Francis turbines. However, for high head-low flow applications, a Pelton turbine is likely to be more efficient than a PAT and no more expensive. PATs are most favorable for medium head sites where practical and cost advantages are in favor of pumps compared to conventional turbines. They require constant discharge and hence are suitable for the regions where continuous supply of water is available round the year. Also, long term water storage is usually not preferred for micro-hydro schemes due to higher construction cost of reservoir. The relation between maximum efficiency in pump and turbine modes proposed by different researchers is given in Table 4.

4. Cavitation analysis of PAT

Cavitation is the phenomenon of formation of vapor pockets or cavities in the interior or on the boundaries of a moving liquid and their subsequent condensation. It occurs in the turbomachinery when the pressure drops, on account of the acceleration of water in the rotor, below the vapor pressure of the water at prevailing temperature. The design, operation and refurbishment of hydraulic turbines, pumps and PATs are strongly related to cavitation phenomena, which may occur in either the rotating runner–impeller or the stationary parts of the machine [66]. It may be detected by carrying out the analysis of structural vibrations, acoustic emissions and measurement of hydrodynamic pressures in the machines [67].

The effects of cavitation are hydraulic (low efficiency due to flow instability) and mechanical (surface damage, noise and vibration). In addition, it may also lead to surface erosion which may vary from a relatively minor amount of pitting after many years of operation to the disastrous failure of large and expensive structures in a relatively short period of time [68]. It is difficult to avoid cavitation in hydro turbines but certainly it can be reduced to an economically acceptable level [69]. In order to reduce it, the available net positive exhaust head for the turbine must be larger than the value required by the machine. Few investigators have studied the performance of PAT under cavitating conditions which is summarized in this section.

Gantar [70] experimentally studied the performance of propeller pump in turbine mode under cavitating conditions and reported that, cavitation characteristics in turbine mode operation are more favorable than in pump mode, therefore, the turbine mode operation requires smaller submergence of the impeller than the pump operation.

Alatorre-Frenk [38] performed number of experiments to study the effects of cavitation number on the efficiency of PAT. It was found that for $\sigma_T > 0.3$, for most of the cases, η_T was in the range of $79 \pm 0.5\%$ with small dispersion due to the oscillations of the system. For $\sigma_T = 0.25$, distinct increase in the efficiency of about 1.5% was observed, which was attributed to the reduction in the skin friction because of the formation of a thin vapor layer on the surface of the rotor. The critical value of cavitation number was found to be 0.25 on account of generation of instabilities in the test rig below this value. The σ_{Tcrit} was found to be lying outside the expected range of 0.16–0.21, which may be due to the difference in size and geometry between the PAT and the conventional turbines as well as due to a larger amount of dissolved and entrained air. The results showed that the cavitation is slightly worse in PATs than in conventional turbines of similar specific speed. In order to evaluate the σ_{Tcrit} values and to formulate a general cavitation theory, it was advocated to carry out both destructive and non-destructive cavitation tests on more PATs, particularly for the large capacity pumps considering the high cost of these tests.

Table 4
Relation between maximum efficiency in pump and turbine modes.

Name of investigators	Relation between pump and PAT BEP
Hancock [34]	$\eta_t = \eta_p \pm 2\%$
Derakhshan and Nourbakhsh [10], Williams [16], Agostenelli and Shafer [25], Childs [33], Isbasoiu et al. [53], Morros et al. [73], Gantar [70]	$\eta_t \approx \eta_p$
Chapallaz et al. [27]	$\eta_t = \eta_p$–(3% to 5%)
Nautiyal et al. [43]	$\eta_t = \eta_p$–8.53%

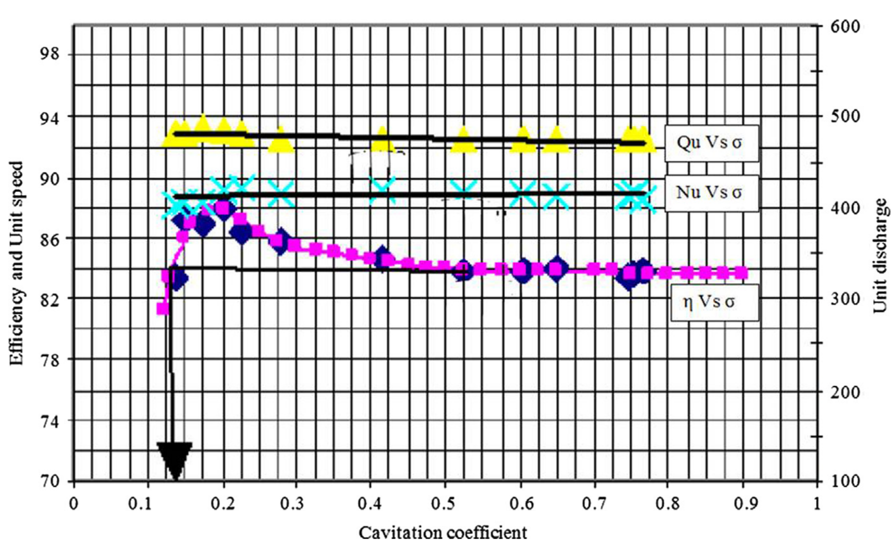

Fig. 9. Variation of efficiency, unit speed and unit discharge with cavitation coefficient.

Singh [44] discussed two criteria of analyzing the cavitation performance in PATs viz. one based on the requirement criterion and the other based on the availability criterion. The requirement criterion was machine dependent; whereas, the availability criterion was depending on the combination of system conditions like draft tube design, turbine settings and machine conditions related to suction blade geometry. To understand the cavitation characteristics of PAT, a non-dimensional Combined Suction Head Number (CSHN) was developed based on the combination of availability criterion and the impeller properties. The CSHN analysis along with the suction specific speed was applied for prediction of pressure near the suction eye of the PAT at different operating speeds, draft tube designs and turbine settings. The minimum value of the CSHN, representing the minimum pressure, was considered as the limiting design factor for the cavitation. From the analysis, the minimum CSHN was found to occur after the BEP (i.e. in the overload region) for all the radial flow PATs and before the BEP (i.e. in the part load region) for mixed flow PAT. The proposed analysis along with Dixon's criterion, which is in accordance with availability conditions, was recommended as the techno-economic criteria to study the cavitation under critical conditions in PATs.

Prasad et al. [11] discussed the suitability of end suction mixed and axial flow pumps as turbines for medium and low head sites. Two pumps were tested in turbine mode to analyze the efficiency and cavitation characteristics under different operating conditions. The experimental setup consisted of high pressure tank, PAT, draft tube, low pressure tank, circulating pump, resorber and vacuum pump. The efficiency test was carried out by keeping the tail race tank open to atmospheric pressure. While during the cavitation test, the tail race tank was connected to a vacuum pump for reducing the pressure at water surface in the low pressure tank. The effects of cavitation coefficient (σ) on the unit speed, unit discharge and PAT efficiency were plotted as shown in Fig. 9. With decrease in σ, initially the efficiency remained constant, then increased followed by sudden drop. However, the unit speed and unit discharge remained almost constant with variation of σ. At BEP, the head and discharge were found to be higher in turbine mode than that in pump mode.

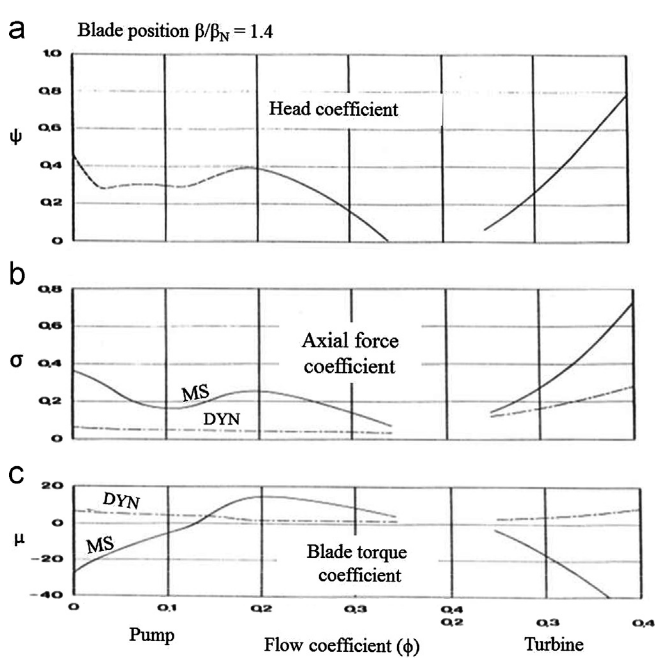

Fig. 10. Variations of (a) head coefficient (b) axial force coefficient and (c) blade torque coefficient with flow coefficient.

It was found that cavitation characteristics in turbine mode operation are more favorable than in pump mode. Also, its effects may be more critical than that in conventional turbines having similar specific speed. In order to evaluate the σ_{Tcrit} values and to develop the generalized cavitation theory for PAT, it was recommended to carry out more detailed investigations, destructive as well as non-destructive, in the wide range of specific speed.

5. Force analysis of PAT

When centrifugal pump is operated as centripetal turbine, the point of maximum efficiency shifts towards a higher head and flow rate than that correspond to pump mode. Hence at BEP, the pump is subjected to higher head and discharge in turbine mode as compared to that in pump mode [11,14,20,71]. Consequently, the flow instabilities and the radial unbalance generated in turbine mode are expected to be different compared to that in pump mode. Many researchers have studied the forces acting on pump in both modes.

Gantar [70] carried out experimental investigations on propeller pump running as turbine. The increase of axial thrust was found to be directly proportional to the head (pressure) coefficient (ψ). In the region of over optimum flows, the axial thrust was observed to be 2–2.5 times greater than that in pump mode operation. For the better performance of the machine, it was recommended to control the load on the axial bearing and eventually, if required, use of large size bearing was suggested. In addition, in turbine mode the torsion moment was found to be approximately 2–3 times greater than that in the pump mode. The variations of head, axial force and blade torque coefficients with flow coefficient in pump and turbine modes are shown in Fig. 10. The performance of PAT under runaway condition was studied. It was observed that the runaway speed was twice the rotational speed at optimum position of the impeller blades and it was decreasing rapidly with the closure of impeller blades.

Barrio et al. [71] used the numerical model to estimate the radial load on the impeller as a function of flow rate in pump and turbine modes. The unsteady flow computations were applied along one blade passage and the resulting radial load was calculated by integration of the instantaneous pressure and shear stress distribution over entire impeller surfaces in each of the time steps. In pump mode, the radial load was found minimum near the design conditions while in turbine mode, it was observed that the magnitude of the radial load increased with the flow rate. Below turbine rated conditions, the magnitude of the total radial load (steady and unsteady components) was found to be lower than the maximum total load in the pump mode. In contrast, significant rise in load was observed above rated conditions. It was concluded that, the mechanical design of the machine and shaft bearings must be carefully undertaken for the smooth functioning of pump in turbine mode. The average magnitude of the radial load on the impeller in pump and turbine modes at different flow rates is shown in Fig. 11.

Fernandez et al. [72] estimated the radial force acting on the impeller of a centrifugal pump in turbine mode by performing steady and unsteady numerical simulations. The variation in average radial force was found fairly linear with the flow rate in turbine mode. The force vector was rotating in the direction of impeller rotation during the blade passage period. With increase in flow rate, the locus of the force vector followed a quasi-elliptical path with rising amplitude. The maximum magnitude of force was obtained when the trailing edge of one of the blades (pressure side) was located at 3° downstream of the tip of the tongue. In the earlier studies the authors reported that the average radial force in turbine mode is lower than that in pump mode [20]. However, the

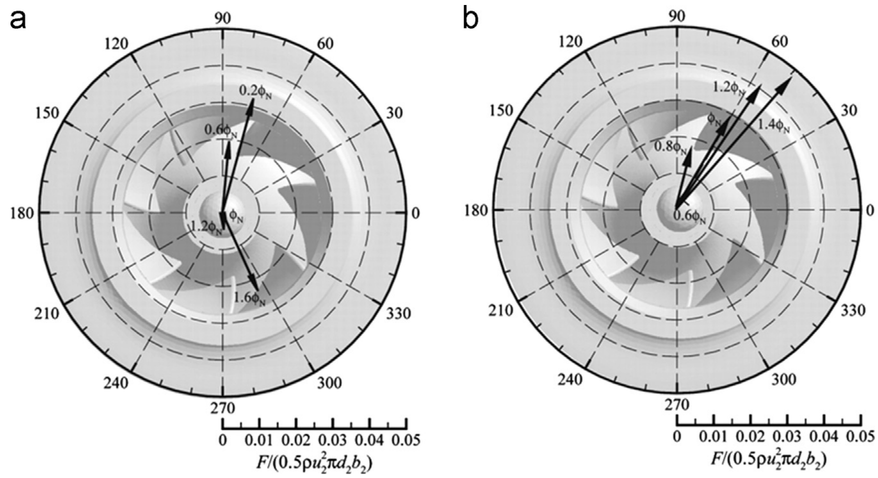

Fig. 11. Average radial load on the impeller as a function of flow rate in (a) pump and (b) turbine modes.

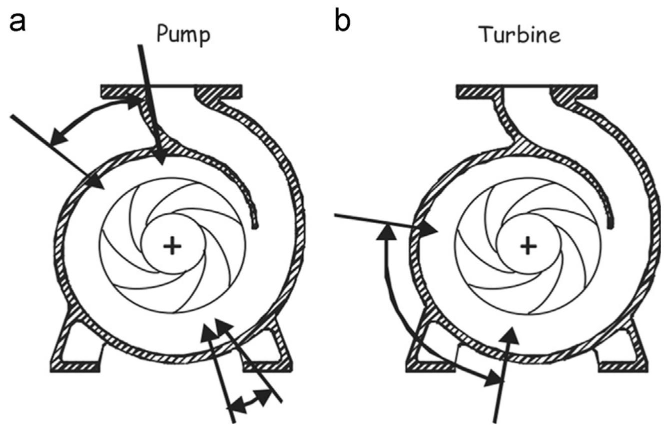

Fig. 12. Locations of radial thrust concentration rate in (a) pump and (b) turbine modes.

results of the current research demonstrated that the unsteady effects due to blade–tongue interaction might be quite significant particularly at higher flow rates in turbine mode and hence same could not be neglected. Fig. 12 shows the areas of radial thrust concentration in pump and turbine modes.

Morros et al. [73] presented the numerical simulation of a radial flow centrifugal pump operating as turbine and validated the results with corresponding experimental data. The volute tongue led to peripheral restriction to the flow which resulted in non-axisymmetric entry of fluid in the impeller. It resulted into substantial unsteady fluctuations over the blade loadings along with imbalanced axial thrusts over the machine bearings. It was emphasized that the unsteady oscillations of these forces may produce severe dynamic loads subjected to high levels of unsteadiness particularly when working beyond the nominal intervals. These fluctuations may go up to 25% of the steady component and result in fatigue failure of the mechanical components. To satisfy all the needs of small hydraulic resources, it was recommended to use PAT for few hours under nominal conditions.

6. Loss distribution in PAT

When pump is operating in turbine mode the direction of flow is reversed; hence, the pattern of loss distribution may not be same as in the pump mode. To improve the performance of PAT, one of the important factors is to identify the causes for losses that may occur in turbine mode. Few investigators have studied the losses taking place in PAT based on theoretical and numerical investigations.

Chapallaz et al. [27] reported that the relationship between the pump and turbine mode performance is not the same for all types and sizes of pumps. But it depends on the flow pattern through the machine, expressed by the specific speed, and the losses incurred, expressed by the efficiency of the machine. As the fluid passes through the pump impeller, it is subjected to friction losses, shock losses and leakage losses in both the modes. Due to these losses, the ideal energy transfer from the rotating impeller to the fluid, as expressed by the Euler equation, is not achieved in pump mode. The magnitude of these losses in the turbine mode is not exactly the same as in the pump mode as shown in Fig. 13. When PAT is operated at optimum flow conditions, an increased pressure ought to act on the PAT resulting in friction and shock losses on the fluid. In addition, small quantity of fluid by-pass the impeller and do not contribute to the energy transfer. To maintain optimum head and flow conditions, the flow approaching the PAT must be increased to compensate for the by-pass flow.

Rawal and Kshirsagar [14] carried out numerical simulation of PAT to determine the losses in the different components. The regions of losses were identified as stationary casing, rotating impeller and the portion of the draft tube up to the location of the measurement point. The hydraulic efficiency of PAT was estimated by considering hydraulic losses in different flow passages. Table 5 shows the distribution of losses in different domains at different discharge as obtained from CFD. In pump mode, the losses in the impeller were relatively small compared to the losses in the casing; however in case of PAT, the loss distribution was found to be reversed, i.e. in casing the losses were less compared to the impeller. It was revealed that, the draft tube losses mainly depend on the swirl angle and cannot be controlled. The power estimated by CFD analysis was found to be higher than the net power measured experimentally because the power losses in friction, bearing and seals were not considered in CFD simulation. Also, the leakages through the stuffing box and the recirculation behind the impeller were neglected.

Derakhshan and Nourbakhsh [58] analytically determined the various losses occurring in the pump running in turbine mode viz. volute power losses, leakage losses, kinetic energy losses at the outlet, hydraulic losses and mechanical losses. The net power output from PAT was worked out by subtracting these losses from the gross power.

Yang et al. [74] discussed the hydraulic loss distribution in the different zones of PAT viz. volute, gap between volute and impeller, impeller and outlet pipe. The variations in hydraulic

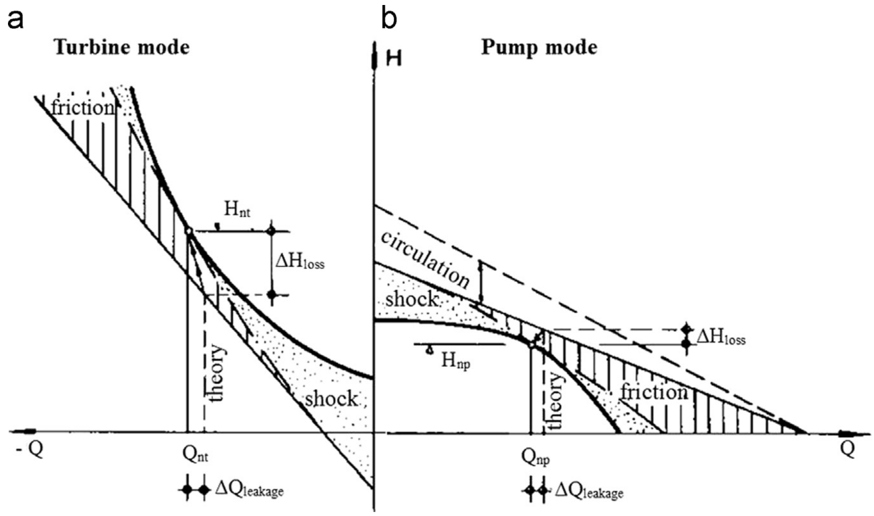

Fig. 13. Hydraulic losses in (a) turbine and (b) pump modes.

Table 5
Loss distribution across various domains of PAT [14].

Parameters	Discharge ratio Q/Q_{BEP}				
	0.6	0.8	1.0	1.2	1.4
Losses in casing (m)	0.59	0.14	0.06	0.03	0.04
Net total head across the impeller (m)	3.61	7.83	12.49	16.90	22.32
Losses in impeller (m)	1.27	1.08	1.34	2.03	3.74
Losses in draft tube (m)	0.97	0.42	0.54	1.10	2.01
Total input head in turbine (m)	6.43	9.46	14.13	20.04	28.10

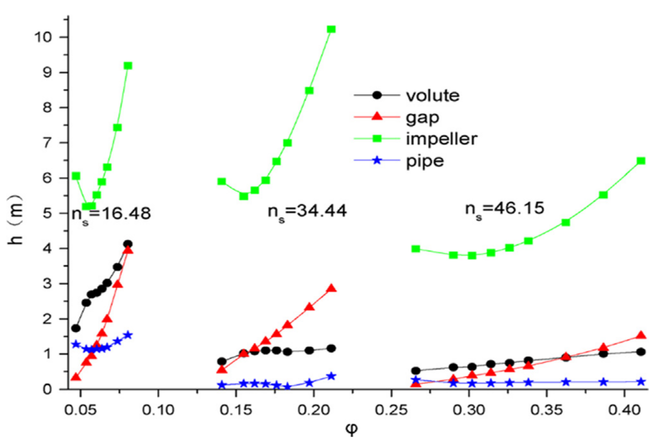

Fig. 14. Hydraulic loss distribution within the investigated PATs.

losses with discharge for three different PATs (specific speeds of 16.48, 34.44 and 46.15) are shown in Fig. 14. It was found that the hydraulic loss within the impeller was dominant compared to other losses and grows with specific speed. The PAT's highest efficiency was reached when the hydraulic loss within the impeller was the smallest. Analysis concluded that the efficiency improvement of PAT should mainly be focused on the optimization of the impeller. A detailed study of the hydraulic loss distribution illustrated that the hydraulic losses in PAT decreases with the decrease of the blade wrap angle. The two main reasons for this decrease were reduction in the length of the impeller flow passage and velocity gradient within the impeller.

Lueneburg and Nelson [26] summarized various factors affecting the total head characteristics as friction losses, absorbed head, shock losses and outlet losses. From the analytical and experimental investigations, it was concluded that the optimum overall efficiency can generally be achieved when the shock losses at inlet to the runner becomes negligible and the absolute velocity at the outlet becomes minimum. The overall efficiency in turbine mode was found to be same or little higher than that in pump mode depending on runner vane angle and nozzle velocity considerations.

7. Performance improvement in PAT

The cost per kilowatt of the energy produced by small hydropower plants is usually higher than that of large hydropower plants. The use of PATs in a wide range of small hydro sites has been gaining importance worldwide in the recent years, but the subject of hydraulic optimization still remained an open research problem. Conventional hydro turbines usually show higher efficiency than the pumps which are designed for the similar operating conditions. On the other hand, the best efficiency of a pump running in turbine mode is almost the same as that in the pump mode [10,16]. Therefore, the application of reverse pumps in larger capacities is not economical. By increasing the overall efficiency of reverse pumps, they can be economically applied in the higher power capacities [75,76]. Many investigators have attempted different techniques to improve the performance of PAT. Lobanoff and Ross [77] reported that, the optimum efficiency of PAT can be generally achieved when the shock losses at the pump impeller tips become nearly zero. Therefore, the modification must be made to the impeller tips of the pump.

Singh et al. [78] studied the effects of presence and absence of the rib situated in the suction eye of the casing, which is provided to break the pre-swirl generated at the inlet of the pump in pump mode. It was mentioned that, if the rib is large it may lead to localized acceleration of the flow, increased hydraulic resistance and breaking of the swirl at PAT outlet. These parameters will change the exit velocity triangles and significantly influence the overall characteristics namely pressure, torque and efficiency characteristics. From the experimental analysis, it was concluded that the absence of casing eye rib, affects the loss patterns of the rib-zone and the internal impeller which resulted in 1.3% rise in maximum efficiency, with a maximum operating efficiency over 82%. Also, the internal swirl study at the draft tube entrance

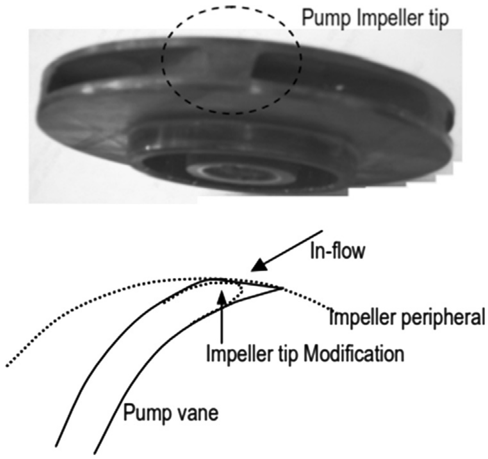

Fig. 15. Modification at the impeller blade tip.

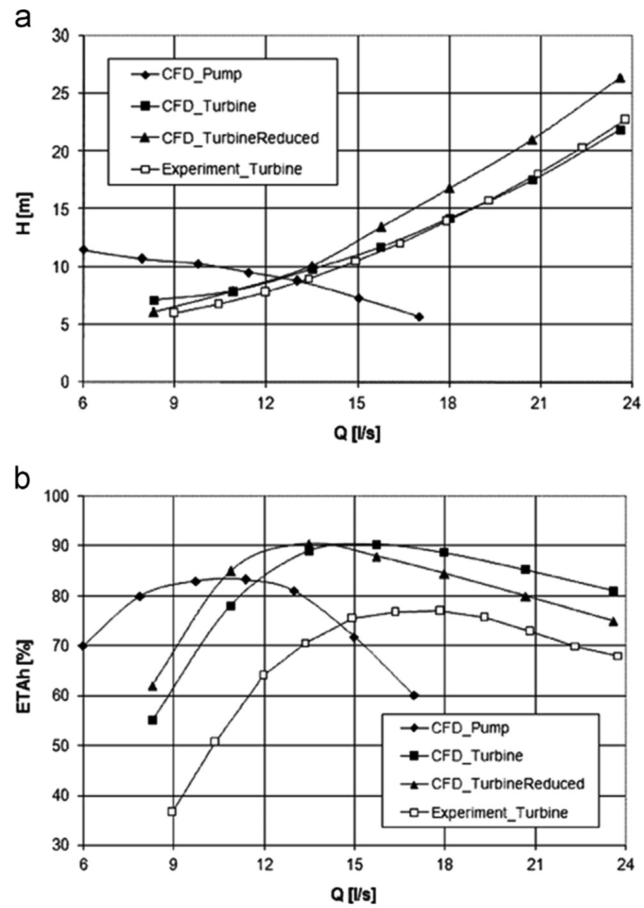

Fig. 16. Comparison of results with full-scale and reduced size impellers (a) head versus discharge curve and (b) efficiency versus discharge curve.

revealed that the absence of rib caused an increase of the rotational momentum of the fluid only in the part-load and overload operating regions. In the rib-zone, the numerical results supported the experimental findings but in the impeller zone the characteristics were differed.

Singh [44] emphasized that the issue of hydraulic optimization is the next stage of research activity in PATs and should be considered at par with the issue of prediction model. Various possibilities of modifying the pump geometry to enhance the performance in turbine mode were demonstrated viz. by rounding of inlet edges of impeller, by modifying the inlet casing rings, by enlarging the suction eye, by removing the casing eye rib. It was revealed that, among the different modifications proposed, the impeller blade rounding was the most beneficial. The modification was applied on eight different centrifugal pumps and found that it has steadily enhanced the overall performance of all the tested PATs with rise in efficiency in the range of 1%–2.5% in the BEP and overload regimes. This modification was first carried out by Lueneberg and Nelson [26] and Cohrs [46] on individual pumps and they stated an efficiency enhancement in the range of 1.5%–2%. Derakhshan et al. [75] used a computer model to analyze the effects of impeller blade rounding on a low specific speed pump.

Suarda et al. [76] experimentally determined the performance of a small centrifugal volute pump running in reverse mode after grinding the inlet ends of the impeller tips to a bullet-nose shape as shown in Fig. 15. The testing was carried out on PAT at the maximum head of the pump by varying the discharge at HuaiKra Thing village. The modifications led to decrease in excessive turbulence and resulted in somewhat improvement in power output and efficiency. However, the modification was suggested only for large capacity pumps as the small pumps are less efficient and the modification could not result in considerable improvement. The efficiency was found to be relatively stable when discharge was varied between rated and maximum capacity.

Williams and Rodrigues [79] discussed the suitability of PAT in place of cross flow turbine for medium head micro-hydropower plants. A 5 kW capacity centrifugal pump working under 22 m head and 2 lps discharge was selected to carryout experimental and numerical investigations in reverse mode. The suction eye of the impeller was enlarged by removing the material and its effects on different zones of PAT were discussed. The modification resulted in huge reduction in the losses in the draft tube which led to reduction of head and rise in efficiency, though the power remained unaltered. Also, the effects of reduction of impeller diameter on PAT were studied and an improvement in PAT efficiency was found. It was observed that, CFD analysis gives useful insight into hydraulic losses; however, for better results need of the improved geometrical model was emphasized.

Derakhshan et al. [75,80] redesigned the blade shapes by gradient based optimization technique, which involves incomplete sensitivities for radial turbomachines to obtain higher efficiency. Optimization program was coupled to FINETURBO V.7, to solve 3D incompressible Navier–Stokes equations and AUTOGRID 5 mesh generator developed by Numeca software [81]. In the next step, rounding was done at blade leading edges as well as at the outer and inner edges of hub and shroud. After each modification, a new impeller was manufactured and tested in the test rig. The rounding of the impeller has reduced the net flow separation loss at the blades and the impeller. However, this may change the inlet velocity triangle and hence shock losses. These modifications led to rise in maximum efficiency in the range of around 3%–5% at rated discharge. The experimental results confirmed the numerical efficiency improvement at all the measured points. Study revealed that, the efficiency of the pump in reverse operation can be improved just by impeller modification.

Sedlar et al. [82] studied the effects of diameter on radial-flow multistage pump working as turbine. Two versions of the impeller were used: the first one with design diameter of 0.25 m and the second one with the diameter reduced to 0.22 m. The flow analysis was carried out at seven flow rates for several rotational speeds in both modes; nevertheless the results for only 1000 rpm were presented. In case of turbine mode, the flow rate was varied from 8.3 to 23.6 lps and the values of peak hydraulic efficiency were found to be 90.5% and 90.1% at optimal flow rates of 14.9 lps and 13.8 lps with full scale and smaller impellers respectively. The comparison of flow patterns inside two impellers showed that,

Table 6
Details of instruments used in experimental setup [83].

Variable	Device	Measurement principle	Range	Accuracy
Inlet head (positive)	Pressure transducr	Inductive+wheatstone bridge	0–2 bar	±0.5% of full scale
Exit head (negative)	Pressure transducr	Inductive+wheatstone bridge	0–1 bar	±1% of full scale
Discharge	Magnetic flow meter	Fraday's magnetic law	0–200 lps	±0.1% of full scale
Torque	Torque sensor	Wheatstone bridge	± 100 Nm	±0.1% of full scale
Speed	Speed sensor	Optical counts		±1 rpm

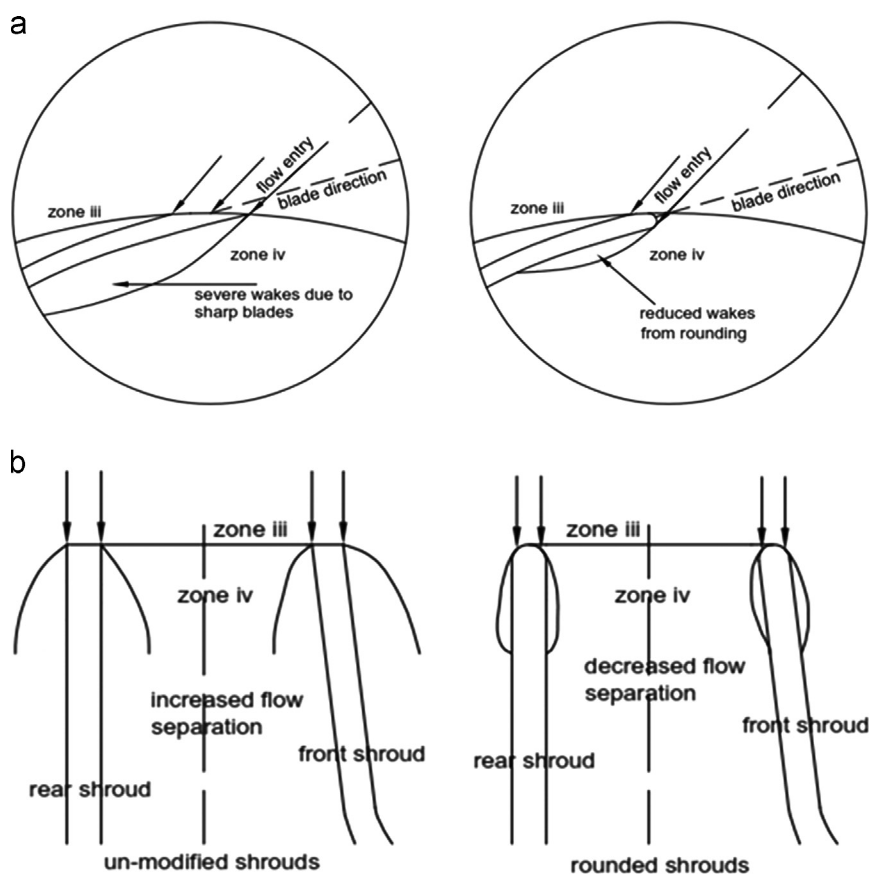

Fig. 17. Effects of impeller rounding at the inlets of (a) blades and (b) shrouds.

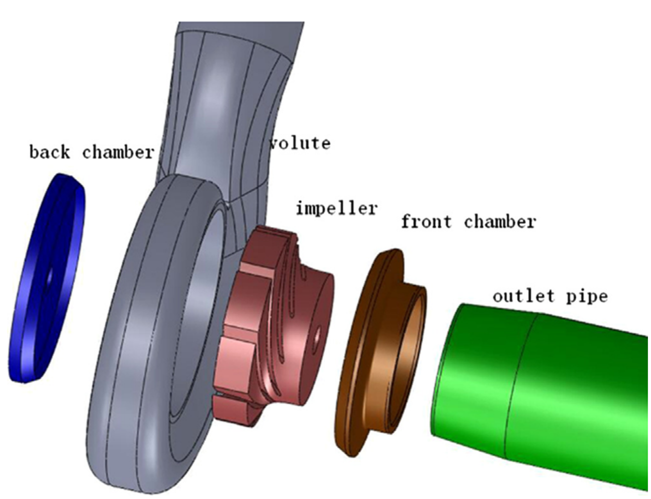

Fig. 18. Different flow domains of PAT.

the flow characteristics were similar in the stator but large global separations were observed inside the impeller passages, which may be the reason for the poor performance with smaller impeller.

The effects of two impellers on head and hydraulic efficiency are shown in Fig. 16. It was found that, the reduction in impeller diameter results in slight shift in optimal discharge towards the left along with slight drop in hydraulic efficiency.

Singh and Nestmann [83] studied the effects of impeller rounding on 9 PATs covering a specific speed range of 20–94.4 rpm (8 radial and 1 mixed flow type). The details of instruments used in experimental setup are given in Table 6. The radius of rounding was kept as half the blade thickness at blade inlets and half the shroud thickness at shroud inlets respectively. Due to these modifications, the system loss coefficient decreased whereas the exit relative flow angle slightly increased for all the PATs. The decrease in system loss coefficient led to drop in net head but the shaft power remained unaltered. Also, the increase of exit relative flow angle resulted in increase in net Euler momentum as well as net head across the PAT. The modifications resulted in reduced wake formation due to bullet shaped rounding at the inlet of the blades and decreased flow separation at the entrance at shrouds as can be seen in the front and side views of the impeller respectively in Fig. 17. As a result, the efficiency was increased for all the PATs (within+2% band) in the part-load, BEP and overload regimes. It was recommended to standardize the impeller rounding effects over wide range of pumps, including

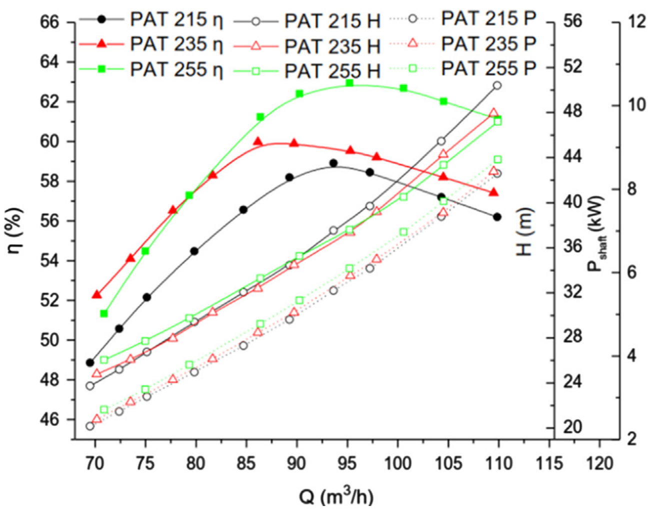

Fig. 19. Performance curves of the three tested impellers.

Table 7
BEP parameters in pump and turbine modes [87].

Items	Impeller dia. (mm)	Q (m^3/h)	H (m)	P_{shaft} (kW)	η (%)	N_{sp}
PAT	255	95.23	39.17	6.15	62.95	15.58
	235	86.14	32.40	4.56	59.98	17.09
	215	93.63	37.52	5.63	58.84	15.96
Pump	255	52.12	18.07	4.19	61.15	20.59
	235	51.71	17.17	3.83	63.08	21.31
	215	49.80	13.75	3.13	59.55	24.71

axial pumps, and the extension of study for reversible pump-turbines.

Sheng et al. [84] applied splitter blade technique to improve the PAT performance which is one of the methods used in flow field optimization and performance enhancement of rotating machinery [85,86]. The negligence of fluid flow in the front and back chambers of pump was having insignificant effect in pump mode but having significant effect in turbine mode [58]. Hence, the flow domain was split into five components namely volute, impeller passage, front and back chambers and outlet pipe as shown in Fig. 18 [74]. Also, to achieve relatively stable inlet and outlet flow conditions, four times of pipe diameter was extended at the entrance and exit of PAT. It was observed that, with the increase of splitter blades head was dropped but efficiency was increased over the entire operating range. Unsteady pressure field analysis revealed that the unsteady pressure field within PAT was improved; however, the variation in shaft power with and without splitter blades was negligible. The values of efficiency, pressure head and shaft power obtained with numerical simulations were found to be higher than the experimental results which may be attributed to the negligence of leakage loss through balancing holes and mechanical loss on account of mechanical seal and bearings.

Yang et al. [74] numerically investigated three PATs covering low, medium and high specific speeds with different blade wrap angles to study the influence of blade wrap angle on PAT. A detailed hydraulic loss distribution and theoretical analysis were performed to investigate the reasons for performance change caused by the blade wrap angle. The results showed that there is an optimal blade wrap angle for a PAT to achieve the highest efficiency and it decreases with an increase in specific speed. Also, the PAT's flow versus head (*Q*–*H*) and flow versus shaft power (*Q*–*P*) curves were decreased and the flow rate at the BEP was

Table 8
Softwares for numerical analysis of turbomachines [91].

Name of software	Country
Fluent	UK and US
CFX	UK and Canada
Fidap	US
Polyflow	Belgium
Phoenix	UK
Star CD	UK
Flow 3d	US
SCRYU	Japan

increased with the decrease of the blade wrap angle. The hydraulic losses within the impeller decreased due to the shortened impeller blade passage and reduced velocity gradient within the impeller flow channel. With the decrease of the blade wrap angle, the slip factor of impeller was decreased; therefore, its theoretical head was also decreased. The decrease in hydraulic losses within the impeller and decrease in the theoretical head were found to be responsible for the poor performance of PAT. The numerically predicted performance curves were compared with the experimental results and found in good agreement.

Sheng et al. [87] carried out an experimental research on single stage centrifugal pump with original impeller (255 mm dia.) and with two trimmed impellers (235 and 215 mm dia.) in pump and turbine modes. Fig. 19 shows the characteristic curves of three tested impellers in both modes and Table 7 lists their BEP parameters. Due to trimming its geometric parameters viz. impeller diameter, blade wrap angle, impeller width and blade angle at inlet were changed. It was found that, PAT's efficiency was decreased with the trimming of impeller diameter. As impeller was trimmed from 255 mm to 215 mm, its efficiency at BEP was dropped by 4.11%, *Q*–*H* curve became increasingly steeper and *Q*–*P* curve moved almost parallel down. The decrease of diameter has shifted the flow rate at the BEP towards left of its performance curve; whereas, the decrease of blade wrap angle and increase of width and blade angles moved its flow rate at the BEP towards the right. Analysis concluded that, the overall enhancement in the performance of PAT with impeller trimming was the cumulative effects of these four geometric parameters.

Among the various techniques attempted by different researchers for performance improvement of PAT, the impeller blade rounding is found to be the most promising technique. However, it is more advantageous with large capacity PATs in view of their higher efficiency compared to small PATs. It is also required to standardize the impeller blade rounding effects over the wide range of PATs.

8. Numerical investigations on PAT

The flow inside hydro machines is highly complex due to rapidly changing flow passage, 3D flow structures involving turbulence, secondary flows, cavitation, etc. Hence, the Navier–Stoke equations for such flow involves unsteady, non-linear, coupled, second order partial differential terms which are very difficult to solve analytically. Traditional approaches for flow analysis in such machines were based on theoretical and experimental studies. These methods have been followed for over the years but applications of theoretical methods are limited to simplified cases and experimental approach is very difficult, time consuming and expensive [88]. In the recent years, CFD started to play a key role for the prediction of the flow through pumps and

turbines having successfully contributed to the enhancement of their design [89].

The applications of CFD in the design and analysis of hydro turbines and pumps started about 30 years ago. The first step coincided with the introduction of the finite element method (FEM) into CFD. In late 90s, numerical codes started to evolve from a pure inviscid physical model assumption into realistic viscous turbulent models. Due to high computer power involved, these models were only applied to study 2D or Quasi-3D Euler solutions. Over the years, the complexity continuously increased in stages: via 3D Euler solutions, to steady Reynolds-averaged Navier–Stokes (RANS) simulations of single blade passage using the finite volume method (FVM), extending to steady simulations of whole machines, until today unsteady RANS equations are solved with advanced turbulence models [85]. In addition, a growing availability of computer power and a progress in accuracy of numerical methods, brought turbomachinery CFD methods from pure research work into the competitive industrial markets [90]. The list of software available in the market for numerical analysis of turbomachines is given in Table 8 [91].

CFD is a promising technique that has been used to accurately predict performance of hydraulic machines directly from geometric data and fluid properties. It has been extensively used in the flow field visualization and optimization of rotating machinery [85,92,93]. One of the major advantages of CFD analysis is that it does not require an actual experimental facility which is quite advantageous particularly for the machines subjected to higher specific speeds which require high flow rates [52]. Yet, it is to be noted that, for useful and correct interpretation of the CFD computed flow fields; a deep knowledge of the physics underlying the flow inside turbomachinery is necessary. Moreover, due to the empirical nature of turbulence models applied in these computations, it is of paramount importance to compare the computed results with experimental results in order to achieve good pump designs [94].

Tamm et al. [95] carried out an analysis of a reverse operating pump using CFD tools, but had no experimental data to verify their results. They recommended further calibration of a CFD results with sound experimental results.

Natanasabapathi and Kshirsagar [96] carried out numerical investigations on PAT using Pro-E solid modeler as modeling software and CFX for the simulation. Across the interface between stationary and rotating components, multiple reference frame (MRF) using frozen rotor interface were defined. Initially, the simulations were carried out with unstructured mesh in which, head drop across the turbine was matched with the experimental values but the deviations were observed in the efficiency at discharges away from BEP. Also, the unstructured grid led to some unlikely results like gain of total pressure, where in actual loss was expected. Then further analysis was done with refinement of the mesh but the results were not inspiring as the errors across the frozen rotor interface remain unaltered. Finally, two rings of structured grid were introduced in between the casing and the runner which were connected by a frozen rotor interface. The results were encouraging and the predicted performance showed a good coincidence with the experimental results in terms of quality and quantity.

Rawal and Kshirsagar [14] carried out experimental and numerical investigations on a mixed flow pump in pump and turbine modes. The entire geometry consisted of casing, impeller and draft tube was modeled and an unstructured tetrahedral mesh was applied. In pump mode, the best efficiency flow rate was 0.100 m^3/s at a head and speed of 8.3 m and 1450 rpm respectively. The numerical model of PAT showed very good characteristics with a maximum operating efficiency of 83.10% at a flow rate of 0.127 m^3/s at 12.48 m head while operating at the same speed.

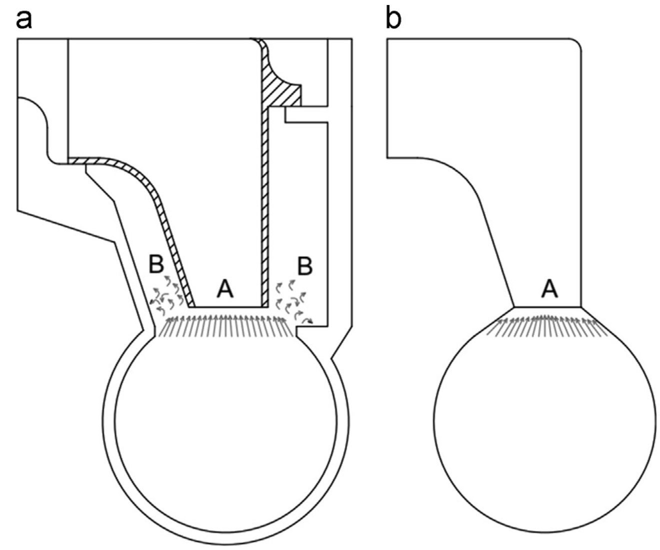

Fig. 20. Volute and impeller interaction: (a) real and (b) CFD-model.

The value of peak efficiency was different at different operating speeds but occurred at a constant Q/ND^3 of 0.4. The maximum efficiencies at 800 rpm, 900 rpm and 1000 rpm were found to be 81.4%, 82.7% and 83.3% respectively. The similarity between experimental and numerical results was satisfactory at flow near BEP flow and above. However, higher deviations were observed at lower flow rates; to minimize the same, use of improved mesh quality, numerical schemes and turbulence models was suggested. In addition, it was recommended to focus on the design of the impeller and the impeller/casing interface for further optimization of PAT.

Shukla [97] carried out CFD analysis of flow field inside the different components of centrifugal PAT using Fluent's MRF model. Flow field inside the volute casing, impeller and draft tube was analyzed at BEP and other values of discharge. CFD results demonstrated satisfactory agreement with manufacturer's data around the BEP in pump mode, but beyond that CFD results showed large deviations. BEP of PAT was obtained at higher values of head and discharge than that of pump. The cavitation analysis showed that the chances of cavitation are minimum at the BEP discharge and increases with the increase of discharge beyond BEP. Correction factors for head and discharge were determined based on the results obtained by CFD technique and compared with the available relation of other studies and found in good agreement. In casing and draft tube, hydraulic losses were observed to be minimum at BEP as compared to other values of discharge.

Derakhshan and Nourbakhsh [58] numerically simulated the centrifugal pump (Ns: 23.5 (m,m^3/s)) in pump and turbine modes. The results were compared with experimental data which showed good agreement in pump mode at BEP as well as at part-load and over-load operating conditions; however, large deviations were found in the turbine mode. At same discharge, the values of head and power obtained by numerical simulations were found to be less compared to experimental results. The variation in the results was due to negligence of flow zone in the space between impeller hub/shroud and casing, as shown in Fig. 20. The effects of geometric simplifications were observed to be higher in turbine mode operation as its effect on downstream flow was more than that on upstream flow.

Sedlar et al. [82] carried out CFD analysis in the wide range of discharge and rotational speeds in pump and turbine modes and reported that CFD analysis provides only hydraulic efficiency; whereas, experimental data signifies the overall efficiency which includes disc and mechanical losses along with hydraulic

efficiency. Hence, the difference of about 13% between the calculated and measured efficiency was considered to be very good.

Nautiyal et al. [98] reviewed the work carried out on the pump running in turbine mode using CFD as a numerical simulation tool and mentioned that CFD is a recent attempt for predicting the performance of PAT. The difficulties associated in grid generation in complex region and its effects on solution convergence were emphasized. It was reported that, the computational analysis is very useful in identifying the losses in turbomachine components like draft tube, impeller and casing. The CFD and experimental results did not match accurately in turbine mode, but it was pointed out that the variation in the results can be decreased by using finer mesh and improved numerical methods and turbulence models. Also, the future work in the field of computational analysis can further improve the prediction of pumps in reverse operation.

Silva et al. [99] presented a full CFD pump computation in pump and turbine modes using an industrial pump as a benchmark test case. In order to obtain a good quality mesh, grid composed of 12 geometrical blocks was used after performing the detailed aspect-ratio and equi-angle adjustments on the mesh domain. The characteristics of three commonly applied approaches for CFD analysis viz. frozen-rotor, mixing-plane and pure unsteady computation were discussed. Considering the existence of a small gap between rotor and tongue the flow was simulated by applying a frozen-rotor approach based on pseudo-unsteady approach in the blade passage, in front of the tongue and the overall turbomachine flow field. It was pointed out that the Spalart–Allmaras model, which is a low-Re turbulence closure, can be used to compute wall bounded flows; however it requires high node density in the near-wall region. Numerical results were compared with the experimental results and the discrepancies were found to be within the expected region for a frozen rotor computation.

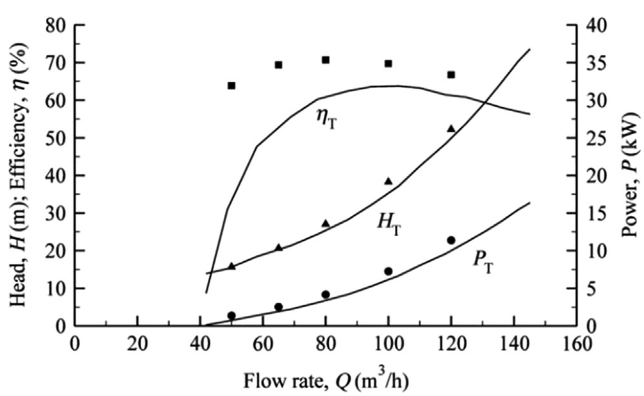

Fig. 21. Performance characteristics of PAT at 1750 rpm. (lines—experimental data; icons—numerical predictions).

Fernandez et al. [72] carried out unsteady CFD analysis of PAT using a sliding mesh technique to consider the effects of blade–tongue interactions on the local flow. The comparison of the numerical results and previously collected experimental results is shown in Fig. 21. Maximum relative errors in total head and average static pressure around the periphery of the impeller were found to be 9% and 5% respectively. The computational model overestimated the magnitudes of efficiency and power, which may be due to negligence of volumetric and disc friction losses as the lateral space between impeller and casing walls were not included in the computational model. At the exit of the impeller tangential velocity component was observed which led to rotation of fluid in the direction of impeller rotation for low flow rates and in the opposite direction for high flow rates.

Barrio et al. [100] carried out internal flow analysis of a centrifugal pump provided with seven twisted backward blades in pump and turbine modes using the commercial code Fluent. The simulations were carried out by solving the full unsteady RANS equations with the finite volume method for the flow rates between 20% and 160% after testing the dependence of the numerical predictions on grid and time step size. Turbulent effects were studied by applying the renormalization group (RNG) k–ε model and standard wall functions were applied near solid boundaries. The numerical results were compared with the experimental measurements which showed typical differences in the range of 3%–5%. In turbine mode, internal recirculation was observed in the impeller at other than rated discharges and large regions of backflow were found in impeller passage. Backflow zones were also observed at 140% flow rate only in the four passageways adjacent to the tongue. Fig. 22 shows contours of static pressure and velocity vectors near the tongue region in turbine mode at different discharges.

Morros et al. [73] presented CFD analysis of PAT and compared the results with experimental data. The fundamental equations viz. mass conservation, momentum conservation and two equation turbulence models (standard k–ε) were discussed and used. The second order implicit scheme was used for the time dependent terms. The study was focused on the basic flow parameters along with energy conversion under unsteady state conditions. A non-axisymmetric flow distribution was observed at the impeller inlet due to the peripheral constraint produced by the volute tongue. Both absolute and relative reference frames were considered to understand the consequences of the tongue interference

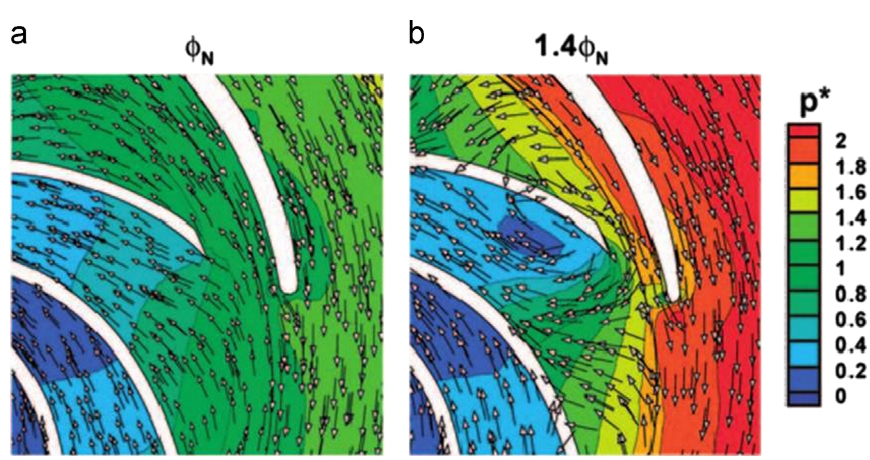

Fig. 22. Contours of static pressure and velocity vectors near the tongue region in turbine mode at (a) 100% and (b) 140% discharge.

on the circumferential blade-to-blade gradients in the impeller passages. The study revealed that PAT can provide acceptable efficiencies which may offset the higher manufacturing costs of specially designed hydro turbines.

The double suction centrifugal pumps are very commonly used when cavitation problems are likely to arise [101]. They are generally preferred in high discharge applications and in situations where the axial forces may put limitation on the use of conventional pumps. Gonzalez et al. [102] studied the performance of double suction centrifugal pump in turbine mode using a CFD simulation based on the sliding mesh and the real movement of the impeller. The computational domain is shown in Fig. 23. The inlet tongue, designed for the pumping mode, was considered as an obstacle at the outlet in turbine mode operation. In spite of the difficulties caused by the geometrical complexity at the exit of the pump, due to the double suction arrangement with a flow splitting effect, relatively high efficiency of 80% was predicted in

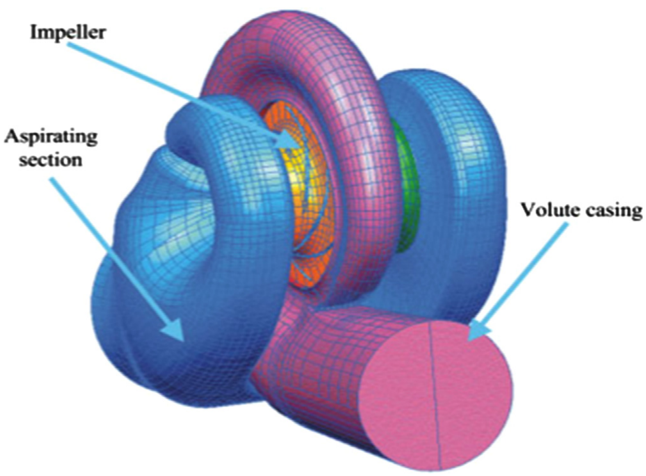

Fig. 23. Computational domain for double suction centrifugal PAT.

Fig. 24. Streamlines in the impeller plane in turbine mode.

the flow coefficient range of 0.3–0.4. The performance was found to be satisfactory in the wide range of discharge, varying from lower to nominal flow rates, except at higher discharge.

Sedlar et al. [82] carried out numerical analysis of middle stage of radial-flow multistage PAT which consisted of the impeller with six blades and the stator with eight channels. The fully unsteady RANS equations with Menter's shear stress transport (SST) turbulence model were solved on block-structured grid consisted of approximately 1.4 million nodes. The mass flow rate along with flow direction specified at the outer periphery of vane less diffuser and average static pressure at the PAT exit were taken as boundary conditions. Two complete stages were analyzed to avoid the effect of boundary conditions on the results; and six interfaces, three rotor–stator interfaces and three stator–rotor ones, were applied between the rotor and stator. It was reported that, the hydraulic efficiency of the multistage PAT can be quite high without making much changes in the manufactured parts. At design condition, the flow was without large separations; but at off-design conditions, highly curved part of stator channel and the region of guide vane–rotor blades interaction were found to be critical. The streamlines in the impeller plane in turbine mode are shown in Fig. 24.

Fecarotta et al. [103] performed the numerical simulation of a multistage pump and showed that the CFD helps in study of turbomachine behavior both in pump and turbine modes. The factors affecting the reliability of the CFD solution were summarized viz. steady or transient analysis, number of elements considered for the analysis, etc. To identify the minimum number of grid elements required, several simulations were performed and found that a 54% reduction in number of elements leads to a decrease in 90% of calculation time and 84% of occupied random access memory (RAM). It was revealed that, the unsteady calculations can simulate the real characteristics of the machine in better manner as they can closely replicate the interactions between the rotor, stator and the hydrodynamic conditions.

Bozorgi et al. [104] simulated an industrial axial flow pump in the reverse mode using RANS equations together with the Spalart–Allmaras turbulence model using Numeca software [81] and validated the results with experimental results. The experimental setup consisted of the axial flow PAT, generator, torque meter, electronic load controller, feed pump, electric motor, venture, butterfly valve, barometers, several pipes and connections; which all were installed on the reservoir. The results showed that the pump can work in a wide range of operation with negligible changes in the efficiency. A first-order uncertainty analysis was carried out using the constant odds combination method, at 95% confidence level in accordance with Moffat [105]. The uncertainties for the head, flow rate and power measurements were found to be $\pm 3.5\%$, $\pm 5.4\%$, $\pm 4.1\%$ respectively. The comparison of numerical and experimental results showed good agreement. Study revealed that, the tested axial PATs can be a good choice

Table 9
Summary of parameters used for CFD analysis of PAT.

Authors	Turbulence model	Pressure–Velocity coupling	Boundary conditions	
			Inlet	Outlet
Rawal and Kshirsagar [14]	Standard k–ε	–	Total pressure	Mass flow rate
Barrio et al. [71]	Standard k–ε	SIMPLE	Total pressure	Static pressure
Fernandez et al. [72]	Standard k–ε	SIMPLE	Velocity inlet	Static pressure
Morros et al. [73]	Standard k–ε	SIMPLEC	Static pressure	Total pressure
Derakhshan et al. [75]	Standard k–ε	–	Mass flow rate	Static pressure
Silva et al. [99]	Spalart–Allmaras	PISO	Total pressure	Static pressure
Barrio et al. [100]	RNG k–ε	SIMPLE	Total pressure	Static pressure
Gonzalez et al. [102]	Standard k–ε	SIMPLEC	Total pressure	Static pressure

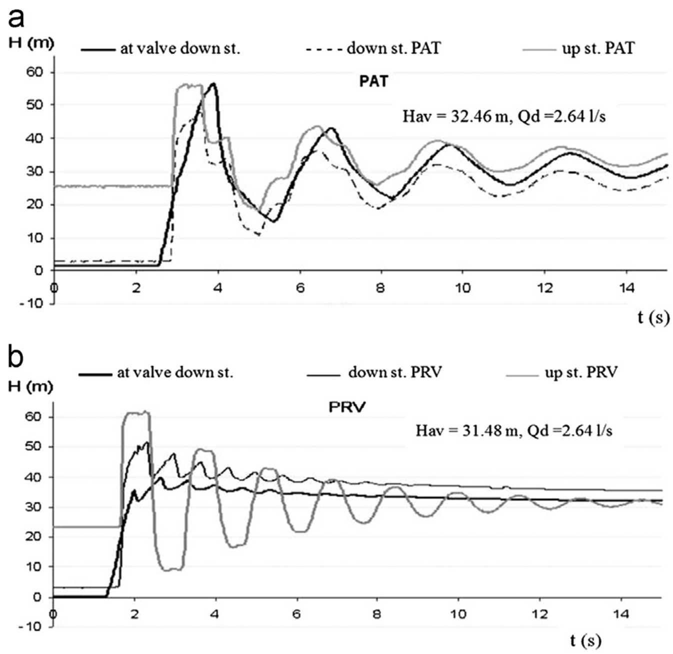

Fig. 25. Dynamic behavior of (a) PRV and (b) PAT: for a fast closure of pipe downstream valve.

for power generation from low-head pico-hydro sites particularly in the developing countries where proper turbines are not easily available.

Several numerical studies on PAT have been reported in the literature, which has proven that URANS equations can provide a reasonable approximation of the general performance of the machine from an engineering point of view with errors typically less than 10% with the experimental data. The reason for such error is difficulties in creation of numerical model; as well as, numerical simulation predicts only the hydraulic performance; whereas, experimental results include mechanical and volumetric losses along with hydraulic losses. The discrepancies between the experimental data and numerical simulation results can be further minimized through improvement in the CFD simulation by use of finer mesh along with advanced numerical schemes and turbulence models. More experience will be needed to realize accurate convergence of CFD results with experimental data. The computational parameters used by different researchers for numerical simulation of PAT are summarized in Table 9.

9. PATs for power generation in water supply systems

Energy saving opportunities in water supply systems has become one of the motivating factors for energy managers and new strategies must be established and applied to harness such energy. In drinking water supply pipe lines, the pressure reducing valves (PRVs) are commonly used as an energy dissipating device to maintain the uniform pressure by localized head loss. The use of PATs in place of PRVs seems to be an attractive solution of controlling the pressure along with energy generation. However, a detailed analysis of PAT under different operating conditions is required to prevent the water supply system from ruptures [103].

Gulich [106] presented the design of centrifugal pump and its performance in turbine mode using mathematical equations. The use of PAT was recommended for energy recuperation in processes where a large amount of fluid energy is dissipated in valves or other throttle devices. The forces exerting in different components during turbine mode were discussed. Theoretical and actual characteristics of a reverse running centrifugal pumps as turbines were illustrated.

Fig. 26. PAT installations in the water piping system at Breech, Germany.

Garcia et al. [63] mentioned that the main pipe in water distribution systems have, in many cases, an excess of static pressure which is usually dissipated by means of intermediate reservoirs, PRVs or any other device that produces the required energy loss. Use of PAT was recommended in such systems to generate electric power up to a capacity of 100 kW. Two cases pertaining to the PAT based water distribution system of Murcia and Elche (in Spain) were considered to compare the prediction methods given by Derakhshan and Nourbakhsh [10], Williams [16], Fernandez et al. [20] and Audisio [107]. The results obtained in two cases under study were found to be very similar and the resulting dimensionless coefficients between points of maximum efficiency were in agreement in both modes with those experimentally obtained by other authors. It was emphasized that the large-scale deployment of PAT is economically viable and acceptable pay-backs can be achieved, which represents an investment opportunity for the companies responsible for water supply network.

Fecarotta et al. [103] studied the performance of multistage pump in a water supply system numerically and experimentally and found that the response of PAT may differ from that of PRV. In case of PRV, the pressure downstream the valve can be controlled by changing the valve position; however, for a PAT (which is normally not furnished with the flow regulating guide vanes) a discharge variation may cause variation in head as well as the pressure waves passing through the runner may cause high efforts and tensions. The difference in dynamic behavior of PAT and PRV for a fast closure of a pipe downstream valve is shown in Fig. 25 [108]. It was emphasized that, under some abnormal conditions PAT may be subjected to water hammer, which may lead to ruptures of the pipelines and loss in efficiency; hence more detailed analysis was recommended under such conditions. The most severe hydro transients occur during extreme operating conditions, such as full-load rejection and turbine stoppages, particularly in hydraulic systems with high head or long pipeline length [109]. Ramos and Almeida [110] studied the behavior of a turbine provided with a guide vane under runaway conditions and water hammer effects on a penstock of a small hydropower plant.

Eight pumps are employed in the Breech electricity recovery system in Germany [111] to reduce the pipeline pressure along with generation of electricity, as these pumps are simultaneously used as turbines for energy recovery. The total installed capacity was 300 kW and the electricity was fed into the grid of the local energy provider by the Regional Water Association. The view of the system is shown in Fig. 26.

PATs also found applications in many fields other than hydropower plants and water piping systems viz. in reverse osmosis

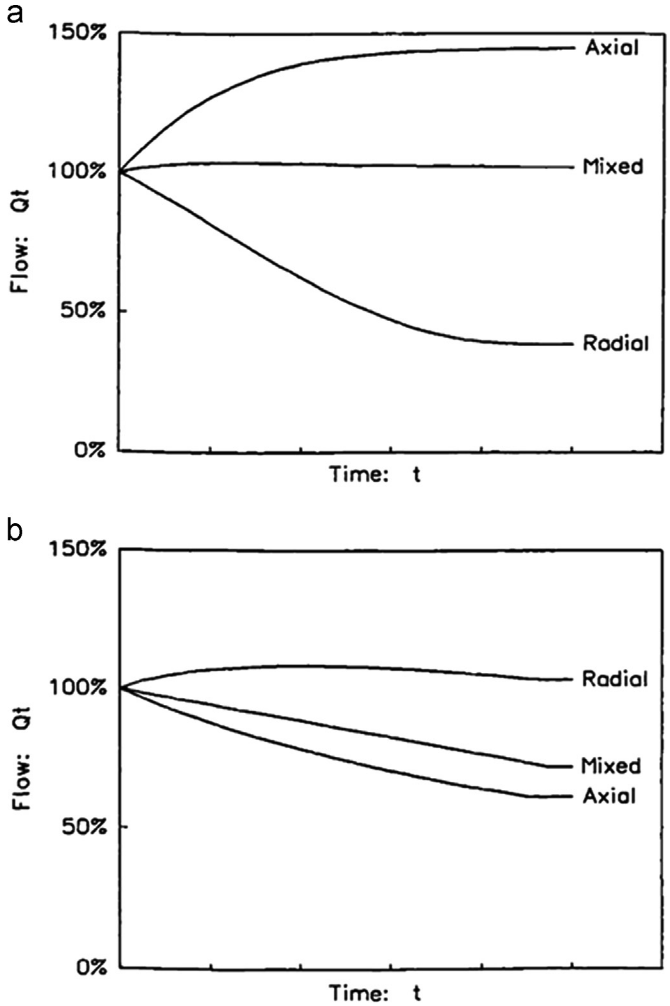

Fig. 27. Change in the flow of PATs in the event of sudden load rejection (a) without brake and (b) when brake is actuated.

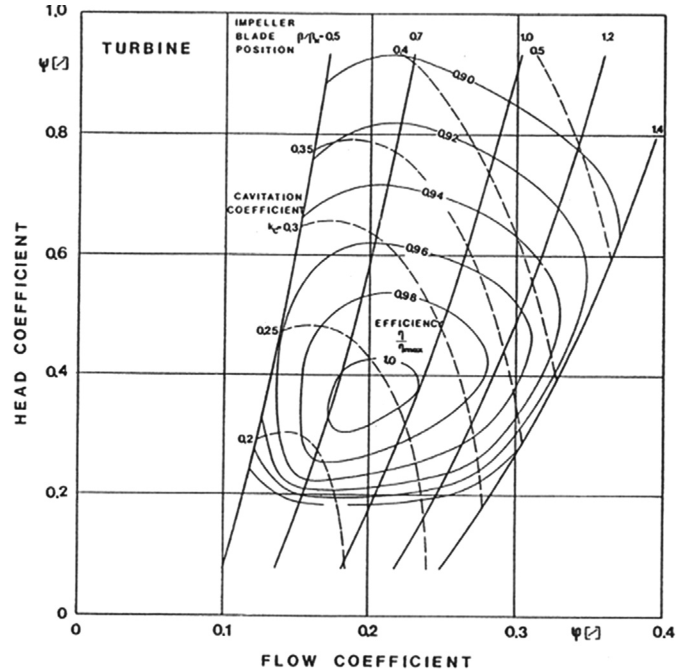

Fig. 28. Hill chart for propeller pump in turbine mode.

systems, in oil and chemical industries (for the decompression processes in gas/liquid separation), in washing/clearing chemical media (for hydrocarbon synthesis by means of hydro-cracking technique), in pressure pipelines inside heating, fuel supply and reclamation systems [112]. However, the information pertaining to such applications is limited and more detailed investigations are required for the implementation of PATs for such applications.

10. Other studies on PAT

Few researchers have done studies on some other issues related to PAT e.g. analysis of steady and transient regimes, controlling of PAT, the water hammer effect, runaway speed, general review, etc. Ramos and Borga [113] carried out the analysis under steady and unsteady conditions based on suter parameters [114] in order to identify the behavior similitude between turbines and pumps, when the pump is operated in turbine mode. Based on the non-dimensional values of speed, discharge, head and torque; different operating conditions of a pump viz. normal pumping/turbining, reverse pumping/turbine, energy dissipation were described. The study was proposed likely to become a pragmatic tool for better understanding about the pumps to generate power especially under runaway conditions. The aim of the study was to get more economical solution to recover some part of the dissipated energy. It was concluded that, the use of PATs allow obtaining a maximum relative efficiency up to 80%, depending on the type of the runner.

In the event of sudden load changes, conventional Pelton and Francis turbines have devices to deflect or by-pass the flow until the spear-valves or guide vanes are slowly adjusted, thus avoiding harmful water-hammer as well as excessive speeds. PATs, on the other hand, have no means to avoid flow and hence speed changes with sudden load variation. PATs can reach the runaway condition in a very short time in the event of sudden load-rejection, as the inertia of typical micro-hydro systems is very small and this abrupt variation of the flow can cause significant pressure surge. A penstock designed to withstand them could have a prohibitive cost because it is usually the most expensive element of micro-hydro schemes. Alatorre-Frenk [38] recommended various techniques of avoiding water hammer effects in PATs viz. arrangement of compressed air tank in the penstock just upstream of the turbine, use of flywheel on the shaft, and provision of rapidly actuating servo-valve with spring loaded brake. The change in discharge of different PATs in the event of sudden load rejection, without and with brake, are shown in Fig. 27.

In PAT, a control system is required to automatically regulate the frequency. The classical governors used for standard turbines are expensive and may not be opted for small hydropower plants. Isbasoiu et al. [53] discussed various methods for controlling the PAT viz. by using the fixed load system, by manual governing of turbine/electric load and by automatic control of ballast loads. The fixed load systems are economical and simple to install but the main drawback is their inflexible characteristic. In case of manual governing of turbine or electric load, an operator is required to adjust the gate valve or to regulate the set of ballast loads depending on variation in the generator voltage and frequency. This method may suffer from large response time and human errors, hence its application was limited to small capacity PATs only. In case of the automatic control method, which can be used with both synchronous or induction generator, the generator load is kept constant by diverting the spare capacity towards ballast loads. The ballast loads should be purely resistive which can be used for various applications like heating, electric drying, water heating, etc.

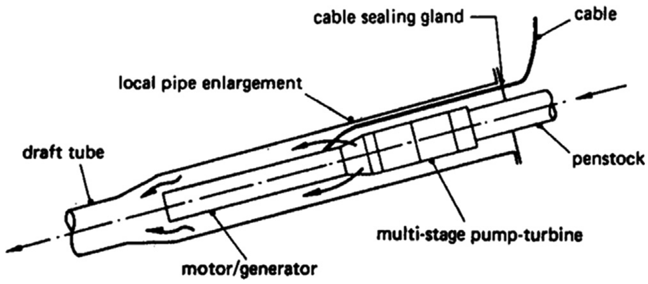

Fig. 29. Submersible pump as turbine in inclined position.

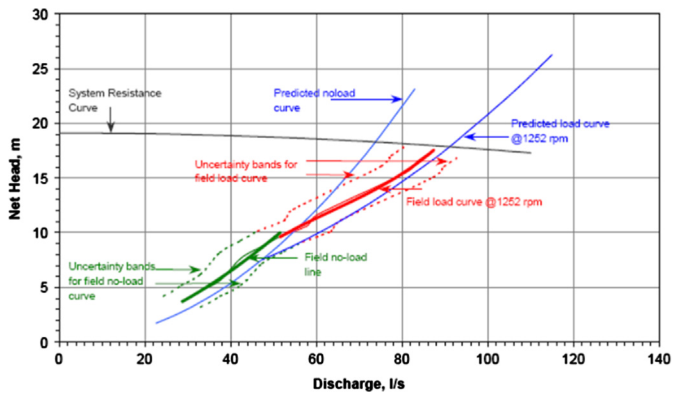

Fig. 30. Comparison of predicted and field hydraulic characteristics.

Gantar [70] described that the propeller pumps which are typically used in irrigation, drainage and in energy object can also be run as energy recovery turbines. The only problem in using these PATs is the suction bell mouth, working as convergent section with small losses in pump mode, which operates as diffuser subjected to higher losses in turbine mode. To utilize the kinetic energy available at the PAT exit, the bell mouth was replaced by carefully designed diffuser with a bend. The hill charts for pump and turbine mode operations were plotted and the region of good efficiency in turbine mode was found to be larger than in the pump mode. This might be due to flexibility of regulating the impeller blades in propeller turbine. The hill chart for turbine mode operation is shown in Fig. 28. The maximum efficiency in turbine mode was found to be approximately equal to that in pump mode; however, BEP parameters were different in two modes. Analysis concluded that, the total cost per installed kW is higher for axial flow PATs in comparison with radial flow PATs; nevertheless, in many cases they can be an efficient way of utilizing the existing energy potentials.

Alatorre-Frenk [38] reported that, in the domain of pumps where impulse principle does not work, very low specific speeds can only be achieved by using multistage pumps. They can be used as turbines, but their cost advantage over impulse turbines may offset by the high cost of multistage pumps compared to that of single-stage pumps. Also, multistage pumps might be costlier compared with small Pelton, Turgo and cross-flow turbines, which can be fabricated in local workshops. Moreover, multistage pumps have higher maintenance costs especially for bearings and seals than Pelton turbines [115].

Giddens et al. [116] experimentally demonstrated the feasibility of operating the conventional submersible pump and motor in the reverse mode as a complete, self-contained, water turbine and induction generator unit. The pump was directly coupled to a single-phase induction motor and when operated in reverse, with the necessary controls, generated an autonomous 220 V, 50 Hz, single-phase supply with an overall system efficiency nearly the same as in the normal pumping mode. The pumping and generating performance of the machine was compared and the principle of the self-excited induction generator was discussed. Possibility of operating submersible PAT in inclined position by housing it in a localized enlargement in the penstock, as indicated in Fig. 29, was also discussed.

Kumar [117] carried out feasibility study on the use of submersible pump in turbine mode for micro-hydropower plants and found that not much information is available on submersible PATs; however, there are no significant technical problems peculiar to use of submersible PATs. The various ways of reducing the capital cost of micro-hydropower plants by installing submersible PAT were summarized as mentioned below:

- Submersible pumps can be installed on the run off river schemes in isolate farms and small villages; where no storage is required, hence it may reduce the civil structure cost.
- It can be run at constant full load with an electronic speed controller; which may avoid the cost of conventional governor.
- It is compact in size as its motor, which will act as a generator in turbine mode, is hermetically sealed and is closely coupled to the pump. This would help to reduce the power house structure cost.
- In the market, submersible pumps are available in standard form, which may further reduce the equipment cost.
- It also provides the flexibility of eliminating the power house structure and hence reduction in capital cost.

Nautiyal et al. [118] reviewed the research work carried out on PAT and categorized into analytical, experimental and CFD investigations. Few modifications carried out by the researchers like impeller modification by grinding of impeller tips, rounding of blade leading edges and hub/shroud inlet edges were discussed. Other applications of PAT e.g. in reverse osmosis and the water distribution system were described. The study revealed that, many efforts have been made by different researchers to predict the performance of pump in turbine mode; however, none of them gives accurate results for complete range of specific speed and hence need of more focused attention was emphasized. Considering the limited studies on impeller modification for the performance improvement of PAT, scope of further research work was recommended in future. CFD studies were found to be inadequate in view of understanding of complex flow physics in PAT and accurate convergence of numerical result with experimental data; however, it was revealed that the further improvement in CFD technique may lead to better understanding of PAT.

11. Case studies on PAT

The performance of any hydropower system in the field may differ from that of the performance studied under laboratory conditions; due to various reasons like generation of artificial head, use of silt and debris free water, scaling effects, etc. Hence, it is utmost important to ascertain the behavior of PAT in the field. In this section, few case studies related to PAT based micro-hydropower plants installed worldwide area are presented.

Maher et al. [119] revealed that, there exists a hydropower potential of at least 3 MW in Kenya, for schemes up to 5 kW in size, which was sufficient to supply basic electricity to 150,000 households in the rural areas of Kenya. In view of this, in December 2001, a community-owned 2.2 kW pico-hydro scheme was commissioned at Thima in Kenya for electrification of 110 low-income rural households. European Commission funded the costs of the penstock pipe and generating equipments; whereas a considerable amount of the total cost including building materials,

Fig. 31. A view of pump as turbine at the site.

distribution cables and house wiring components was met by the consumers. A standard mono-block centrifugal pump was used as a turbine and its motor was used as a generator. The pump impeller was turned down on a lathe to achieve a better operating efficiency. It was revealed that, along with funding further grid extension, the rural electrification levy should be used to inspire the development of pico-hydro and solar home systems mainly for low-income households.

In order to promote small industries in rural communities of Tanzania, United Nations Industrial Development Organization (UNIDO), Environmental Tectonics Corporation (ETC), Netherlands and Tanzania Traditional Energy Development Organization (TaTEDO) installed a PAT based micro-hydropower project of 10 kW capacity in Kinko village of Lushoto District in Tanzania in July 2006 [120]. The produced electricity was supplied to 100 houses in the village to assist in rural development and poverty reduction through improvement of existing and introduction of new livelihood strategies. The maximum PAT efficiency and an overall water to wire efficiency were found to be 80% and 64% respectively. Singh et al. [121] evaluated the field performance of PAT unit for this plant and compared the predicted and field hydraulic characteristics with uncertainty bands for field trials, as shown in Fig. 30. The maximum deviation for the net head, discharge and power output were found to be within 2%–4% only.

A 3-kW micro-hydropower project was installed in Mae Wei village of Tha Song Yang District in Thailand [122] where centrifugal pump was installed in turbine mode. It was installed over 10 days in February 2008 by a team of villagers from Mae Wei, Engineering Studies Program (ESP) students from the Mae La refugee camp, American students from the Institute for Village Studies and Spring Street High School, Border Green Energy Team (BGET) members and Palang Thai. The gross head available at the site was 35 m and water was supplied to the PAT using 4 in. diameter PVC pipe. The pump's induction motor was used as a generator to generate 3-phase 240-volt (delta) electricity which was converted into single-phase 240 V using a capacitor arrangement. A micro-hydro controller was used to dump excess electricity to a resistive ballast load. The generated electricity was utilized to light 16 classrooms, a dormitory and teacher's homes, 10 computers and video equipment.

A team from the University of Itajuba, Brazil [123] conducted a field tests on 45 kW capacity PAT at the Fazenda Boa Esperanca, located on a 211-hectare site in the Minas Gerais province in Brazil to access the technology's appropriateness for practical application. The selections of a pump to be used as turbine and an electric motor as an induction generator (IG) were made in accordance with the technique proposed by Chapallaz et al. [27]. For a net head of 21.8 m and a rated discharge of 0.27 m^3/s, a pump capable of generating a head of 14.48 m and a discharge of 0.212 m^3/s was selected. The system is running since 2007 to the owner's full satisfaction. Also, in order to meet the future electricity demands in an ecological manner, an installation of another PAT/IG set was under planning. A view of pump as turbine at the site is shown in Fig. 31.

The PAT has been installed worldwide in many countries for power generation, in water supply pipe lines and in reverse osmosis systems. The details of some typical PAT installations in the world are given in Table 10.

12. Cost analysis of PAT

The efficiency of PATs is lower than the conventional hydro turbines but their applications are recommended in view of their lower initial and maintenance cost. The initial cost of the machine affects the cost of the hydropower plants only in initial phase of the project; however, the lower efficiency of the machine affects the plant on daily basis. Hence, to justify the use of PAT in mini/micro-hydropower plants as well as to widen its scope of applications, few investigators have applied different approaches viz. comparison of various project investments opportunities based on several financial parameters, evaluation of various renewable energy options, cost analysis of hydropower plant by considering conventional hydro turbine as well as an equivalent PAT as prime movers.

Alatorre-Frenk [38] carried out the cost analysis of micro-hydropower plant based on PAT and conventional hydro turbine. The cost of the scheme was considered as the sum of the initial investments in turbomachinery, penstock, storage reservoir and miscellaneous items. These costs were considered as a function of the size of the scheme, which were further depending on the rated flow, and the lifetime was assumed to be same for both the cases. The yearly operation and maintenance costs were considered as a fixed proportion of the initial investment for all the cases. The characteristics of three most frequently used financial parameters for evaluation and comparison of project investments viz. the net present value (NPV), the benefit/cost ratio (BCR) and the internal rate of return (IRR) were discussed and the use of BCR was recommended for the analysis. The parallel and intermittent operations of PAT were compared with the conventional turbines. It was found that, in spite of the lack of flow control devices in PATs, usually the large reduction in cost makes them more economical than conventional machines. Also, from water–hammer considerations, it was recommended to use an external device to damp the pressure fluctuations instead of using a very thick-walled penstock which led to additional cost saving.

Maher et al. [119] carried out a cost comparison of different off-grid electricity generation options viz. PAT based pico-hydro plant, the solar home system and the battery system in a rural area of Kenya. The annual life cycle cost (ALCC) and cost per kWh were worked out for different options by considering the installation cost, life of the system and annual maintenance and operation costs. The details of cost analysis are given in Table 11. The total cost of 2.2 kW pico-hydropower project was worked out as US$ 7865; which included the costs of civil works (3%); PVC penstock (8%); turbine, generator, controller and protection (22%);

Table 10
Worldwide installations of PAT.

Location	Capacity of plant	Year of installation
Laos, Xiagnabouli province, South East Asia [124]	2 kW	2008
Thima, Kenya [119]	2.2 kW	2001
Mae Wei village, Thailand [122]	3 kW	2008
Barnacre, North-west of England [18]	3.5 kW[a]	1996
West Java, Indonesia [125]	4.5 kW	1992
Kinko village, Lushoto District, Tanzania [121]	10 kW	2006
Fazenda Boa Esperanca, Brazil [123]	45 kW	2007
Ambootia Micro-Hydro Project, Darjeeling, India [44]	50 kW	2004
British Columbia, Canada [126]	200 kW	–
Breech, Germany [111]	300 kW[a] (8 units)	2006
Vysni Lhoty power station, Czech Republic [111]	332 kW ($1\times90+1\times110+1\times132$)	2008

[a] In the water supply system.

Table 11
Cost comparison of off-grid options of household electrification in Kenya [119].

Type of system	Installed cost per household (US$)	Life of system (years)	Annual maintenance and operation costs (US$)	Life cycle cost (US$)	Energy per household over lifetime (kWh)	Average cost (US$ per kWh)
Amorphous silicon, 12 Wp	200	10	35	480	271	1.77
Crystalline, 20 Wp	330	20	35	960	879	1.09
Pico-hydro, 16 W average	81	20	12	321	2102	0.15
Auto battery only, 50 Ah	70	2	33	33	31.2	4.36

distribution system cable, house wiring and energy saving bulbs (43%); house-wiring labor (5%) and costs for design, delivery and project management (19%). From the analysis, a PAT based pico-hydro plant was found to be more cost effective, at less than half the installed cost per household, than an equivalent solar power system. The cost per kWh was worked out to be less than 15% of the cheapest solar home system, which had put it within the reach of most low-income households.

Motwani [127] carried out an ALCC analysis for 3 kW capacity micro-hydropower plant, by considering PAT and an equivalent Francis turbine as prime mover, based on initial cost of the project (Co), capital recovery factor (CRF) and annual expenses (Ac). For the analysis, only initial cost of machine was considered to evaluate the initial cost of the project assuming that the costs of civil works, building and miscellaneous items would be same for both the cases. The CRF was estimated considering 12% annual discount rate (d); and the equipment life (L) of PAT and Francis turbines were taken as 10 and 25 years respectively based on the market survey. As annual expenses, operation cost (@ 5% of initial cost) and maintenance cost (@ 10% of initial cost) were considered; assuming that the costs of manpower and miscellaneous items would be same for both the cases. Based on the analysis, the ALCC and the cost of electricity generated per kWh were found to be 85% and 80% less for PAT compared to Francis turbine, which has justified the use of PAT in place of Francis turbine for their case study. The ALCC and CRF were worked out using following equations.

$$\mathrm{ALCC} = (\mathrm{Co} \times \mathrm{CRF}) + \mathrm{Ac} \tag{15}$$

$$\mathrm{CRF} = \frac{d(1+d)^L}{(1+d)^L - 1} \tag{16}$$

Chuenchooklin [128] presented the cost analysis for pico-hydropower plant for a farming village at the upstream of Wangthong Watershed in Thailand where pump was installed as turbine. The actual gross power produced was 1.116 kW with the net head of 10.74 m and flow rate of 21 lps at maximum system efficiency of 81.5%. The construction cost of the project was approximately US$ 4000 (45% for pipe systems, 37% for control and electricity systems and 18% for pump and turbine systems). Based on the overall electricity consumption of 8760 kWh per year and electricity charges of US 0.75 cent per kWh, the economic recovery period was estimated as 6 years. The results showed that the produced electricity was enough for the indoor electrical appliances such as electric light and some house-ware appliances. It was recommended to install PAT based pico/micro/mini-hydro-power plants in larger farming villages where the higher head and larger flow rate may be available depending on the topography characteristics.

Arriaga [124] carried out the cost analysis of 2 kW capacity project in the Xiagnabouli province in the Lao People's Democratic Republic for isolated communities (40–500 people). Three options were considered for the analysis viz. power generation from hydro resources using PAT or Vietnamese turbine and solar energy using photo voltaic (PV) panels; and the prices were taken from the market of Vientiane. The total cost was comprised of costs of energy generation equipments (EGEs), civil works (CWs) and energy distribution. The EGE included PAT, induction generator controller (IGC) and related electrical equipments. The CW costs involved the weir, canal, fore bay, penstock and PAT foundation material, excluding the costs for labor and equipment transportation. Table 12 summarizes the EGE and CW costs for the three options for power generation. The CW costs were maintained equal in both hydropower cases for ease of comparison. From the analysis, the lowest installation cost was found in case of PAT while the PV approach was subjected to the highest investment cost. Study revealed that, the proposed PAT based hydropower project can provide a long-term reproducible system for communities where pico-hydro propeller turbines were inadequate and proper turbines were quite expensive.

Table 12
Energy generation equipment and civil works costs [124].

Equipment	Capacity	EGE (US$)	CW+Transm. line (US$)	EGE+CW (US$)	EGE (US $/kW)	EGE+CW (US$/kW)
Proposed PAT	2.0 kW	2545	3665	6210	1275	3105
Vietnamese turbine	2.0 kW	4800	3665	8465	2400	4235
PV panels	2.0 kW_p	9000	–	9000	4500	4500

Among the various techniques used by researchers for the cost analysis viz. NPV, BCR, IRR, ALCC; the use of BCR was found to be most promising and it is recommended for the further analysis. For rural and remote area, PAT based pico/micro-hydropower plant was found to be more promising than an equivalent solar thermal, solar photovoltaic or battery systems.

13. Market status of PAT

The concept of using a pump in turbine mode has been recognized by pump manufacturers for many years and within the water supply industry this concept has been implemented to a limited degree to generate power in regions where hydro turbine installation is costly affair. It has been noticed by water suppliers, operators of small hydropower plants and pump manufacturers that PAT is an efficient way of generating energy as well as recovering energy and contributing to energy savings. Worldwide, many pump manufacturers have carried out research on PAT and supplied different types of pumps for power generation in hydropower plants, water supply system, etc.

KSB Aktiengesellschaft is one of a group of pump manufacturers that is active in investing resources in PATs and the company has achieved tremendous success with its solutions in several parts of the world. KSB has already supplied the pumps running as turbines for various applications like small hydropower plants (< 10 MW), major water transport systems, reverse osmosis and industrial plants where the technology can be implemented as an alternative to throttling devices. KSB has been active in supplying volute casing and ring-section pumps for PATs over several years, mostly in the small hydropower market [21].

Andritz Hydro [129] is a global supplier of electro-mechanical systems and services (water to wire) for hydropower plants and has more than 170 years of accumulated experience in turbine design. Their single-stage and multistage centrifugal pumps are used as mini-turbines for different applications e.g. as recovery turbines in pulp and paper mills, in small hydropower plants and to supply energy to mountain refuges and forest lodges. They offer two different series pumps viz. ACT and FPT suitable for turbine mode operation. The ACT Series is characterized by an open impeller and wear resistant design which can handle not only drinking water but also residual and waste water, as well as pulp suspensions. The ACT series PATs are suitable for head, discharge and power up to 80 m, 0.8 m^3/s and 250 kW respectively; whereas corresponding values for FPT series PATs are 80 m, 6 m^3/s and 2 MW respectively.

Kirloskar Brothers Ltd. (KBL) [96] is a major player in the manufacturing and supply of hydraulic machinery. The company's core products are pumps, turbines and valves and are used in irrigation schemes, power sector, process plants and domestic applications. KBL has its product range in hydropower sector and is supplying hydro turbines in mini/micro-range. It was mentioned that, interest on pumps operating in turbine mode is growing for micro-hydropower projects, particularly in the range of 10–100 kW, and the pumps could be used most efficiently and economically for this power range.

The pump manufacturers normally do not provide the performance curves of their pump working in turbine mode, which are necessary to select the correct PAT for the hydropower plants [103]. Worldwide, the focus of the pump companies as well as the researchers has been to develop accurate prediction methods for the turbine mode operation of wide range of centrifugal pumps. In spite of extensive efforts by different researchers as reported by Derakhshan and Nourbakhsh [10], Singh and Nestmann [48], Williams [59] the accuracy of these methods has remained uncertain for all pumps with different specific speeds and capacities. Therefore, it is highly desirable to encourage pump manufacturers to test at least some of their pumps also in turbine mode, which may further widen their markets and contribute to better utilization of available SHP potential [112]. Also, they may produce two impellers for their centrifugal pumps: one for pump mode and another for turbine made [75].

14. Limitations and recommendations

The most critical step in a PAT technology is the selection of the most appropriate pump according to site conditions and the precise prediction of the turbine mode performance [44]. Many researchers have developed the relations for the selection of pump to be used as turbine based on theoretical, experimental and numerical investigations. The results of the same are encouraging.

The hydropower plants are subjected to two kinds of variations in their operating conditions; viz. short term, due to changes in load demand, and seasonal, due to changes in the available head and discharge. In conventional hydro turbines, the rotating speed is maintained constant against variable power demand controlling the discharge by changing the guide vane positions. Since PAT does not have guide vanes, same may be considered as a turbine with full guide vane opening and hence its speed varies according to the varying power output. This may lead to instabilities in PAT at part load and results in poor part load efficiency, which is one of the major issues impeding the PAT technology. The performance curves for different turbines are shown in Fig. 32 [112]. It can be seen that, the conventional turbines have wide operating range between 20% and 90% of the discharge, whereas the PAT works with higher efficiency in the discharge range of only around 80% to 100%; hence, their applications are recommended at the maximum attainable efficiency for fixed load applications, close to full load operation.

To take care of seasonal flow variations, the PATs can be designed for the minimum annual flow rate or the better option could be to run several PATs in parallel to achieve good performance at part load. If two or more PATs are operated in parallel, they can be switched on and off according to the available flow [38]. The part-load problem was effectively solved by synchronizing three PATs on a single shaft in the tea garden in Darjeeling, India [44]. Parallel operation has proven to be more cost-effective than a single conventional hydraulic turbine of comparable capacity [130]: the limit for this advantage is five PATs in parallel according to Fraser & Associates [131], seven according to Nicholas [132] and Hochreutiner [133]. In addition, PATs can also be installed in parallel with conventional turbines, the fine-tuning being given by the later. Running several PATs in parallel requires minimum control; however, this type of arrangement may diminish the low cost advantage of the PAT over the use of a single turbine in some typical cases [21]. Hence, it requires more detailed investigations while deciding the number of PATs to be installed for a particular site.

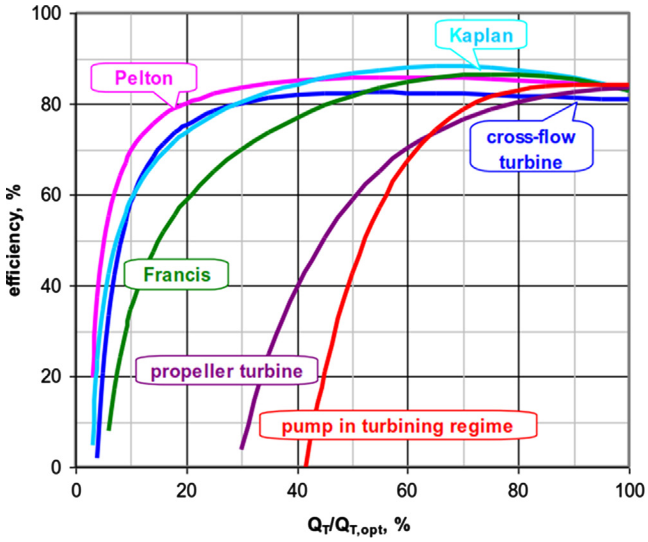

Fig. 32. Performance characteristics of different turbines and PAT.

Another technique to accommodate flow variations with a fixed geometry PATs is to store water in a reservoir and to release it intermittently. As compared with parallel operation, intermittent operation is more efficient and simpler to operate, because it uses all the available water and its operation is automatic. Furthermore, it uses a larger machine that will usually be more efficient and cheaper than several smaller machines. However, the intermittent operation does not have the advantages of parallel operation, viz. the possibility of part-load operation during maintenance and the possibility of installing some of the PATs at a later stage, to reduce the initial cost [38].

Some researchers suggested that the upper limit of application of PATs is established by their market availability 'off the shelf', i.e. around 100 kW as mentioned by Schmiedl [37] or 250 kW by Engeda and Rautenberg [134]. However, other authors affirmed that, even if PATs are to be manufactured for a typical applications, they continue to be more economical than conventional turbines up to a much higher threshold viz. Lawrence [135] mentioned 1.5 MW; Garay [136] 2 MW; Grant and Barn [137] reported about several MW, with various 500 kW machines in parallel; and Hochreutiner [133] described a Swiss hydropower station with seven numbers of 931 kW PATs in parallel.

Hydroelectric power plants are often subjected to run at off-design operation in order to satisfy the fluctuating load demands which would naturally require better performances at part loads. Many efforts have been put by different researchers for improving the performance of conventional hydro turbines under part load conditions viz. by introducing the axial jet from the center of runner of Francis turbine [138], by providing the vane in the bend of the draft tube [139], by placing the guide vanes in hydro turbines [140,141], etc. Similar techniques can also be applied in PATs to improve its part load performance which may widen its scope of applications in mini/micro-hydropower plants. Steller et al. [112] summarized the main research directions in the field of PAT technology viz. identification of home manufactured impeller pumps suitable for energy generation, determination of performance characteristics for specified series of pumps and establishing general relationships between BEP parameters in pump and turbine modes and low-cost modifications of the flow part geometry for enhancement of performance of PAT.

The current practice in the design of draft tube for PAT is to follow the standard available for conventional hydro turbines (IS 5496 [142]) as the standard for the design of draft tube for PAT is not available. However, in certain cases, positive pressure has been observed at the exit of draft tube of PAT; which has recommended the use of constant area pipe at PAT exit in place of diverging shape draft tube. Researchers have either used diverging draft tube [43,96,78,76,11,119] or constant area draft tube [22,10,48,123,143]. Considering the fact that, the behavior of PAT may differ from that of conventional reaction turbines as well as economy involved in fabrication of constant area draft tube more detailed experiments were recommended for the optimization of draft tube [127,143]. Another issue of interest would be the economic and technical comparison between hydraulic ram pumps and pump-PAT sets, aimed at proposing an application boundary between both technologies [38].

15. Conclusions

Pumps are available in the wide range of head and discharge and found wide applications in industries, commercial as well as domestic sectors. They are subjected to various advantages compared to conventional hydro turbines viz. low cost, less complexity, mass production, availability for a wide range of heads and flows, short delivery time, availability in a large number of standard sizes, ease of availability of spare parts, easy installation, etc. It has been found that, technically any type of pump viz. axial flow, mixed flow, radial flow, double suction as well as multistage pumps can be used in turbine mode for power generation. However, from techno-economic considerations use of single stage end suction centrifugal pump, working in the range of low to medium head, is recommended by most of the researchers [19,20,26,43,48,50,51,58,71,76].

In the present study, the different turbines suitable for micro-hydropower plants are explored. The historical background and the current trends in PAT technology are reviewed. The review of literature revealed that lots of theoretical, experimental and numerical work has been done by many researchers on PAT for selection of pump running in turbine mode, cavitation analysis, various methods of performance improvement, force analysis, loss distribution, cost analysis of hydropower plant with conventional hydro turbine and PAT, applications of PAT in water supply systems, etc.

Many researchers have presented theoretical and empirical correlations for prediction of PAT performance from pump characteristics; however, the results obtained with these methods are not reliable for all the pumps subjected to wide range of specific speeds and capacities. Mostly, the maximum efficiency in turbine mode was reported to be same or somewhat less than that in pump mode. The cavitation characteristic in turbine mode operation was found to be more favorable than in pump mode. Also, its effects may be more critical than that in conventional turbines having similar specific speed. To develop the generalized cavitation theory for PAT, it was recommended to carry out more detailed investigations, destructive as well as non-destructive, in the wide range of specific speed. The studies on force analysis revealed that, the volute tongue causes peripheral restriction to the flow and hence non-axisymmetric entry of fluid in the impeller which results in relatively higher axial thrust in turbine mode compared to pump mode.

Among the various techniques attempted by different researchers for the performance enhancement of PAT, the impeller blade rounding is found to be most promising and it is recommended to standardize the impeller blade rounding effects over the wide range of PATs. Several numerical studies on PAT have been reported in the literature, which has proven that URANS equations can provide a reasonable approximation of the general performance of the machine from an engineering point of view with errors typically less than 10% with the experimental data. The

discrepancies can be further minimized through use of the improved computational model, finer mesh along with advanced numerical schemes and turbulence models. Among the various techniques used by researchers for the cost analysis viz. NPV, BCR, IRR, ALCC; the use of BCR was found to be most promising and it is recommended for the further analysis.

The efficiency of PAT is usually found to be lower than that of conventional hydro turbines; however, efficiency is not the primary selection criterion for such machine and its operation is recommended around maximum efficiency point. Also, use of PAT in the range of 1–500 kW may lead to capital payback period of the order of 2 years or even less which is considerably less than that with a conventional turbines. Many efforts have been put by different researchers for improving the performance of conventional hydro turbines under part load conditions viz. by introducing the axial jet from the center of runner of Francis turbine, by providing the vane in the bend of the draft tube, by placing the guide vanes in micro-Kaplan turbine, etc. Similar techniques may be applied in PATs to improve its part load performance which may widen its scope of applications.

The future work for further improvement of PAT performance will be focused on generalized selection criteria, cavitation analysis, modifications in the draft tube, low cost modifications in the impeller, studies on water hammer, fluid structure interaction, etc. The findings of the present study will provide comprehensive details of the research carried out so far, the current trends and the future scope for the development of pump running in turbine mode to the researchers working in the similar area.

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